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Technical Report

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OPTIMAL TUNING OF HEAVY EQUIPMENT MOTION CONTROLLERS

ABSTRACT The objective of this work is the demonstration of feasibility of computerized optimal tuning of electronic motion controllers for mobile heavy equipment end-effectors. A backhoe yaw mode position control system was selected as the application for the purpose of proof of concept. The control algorithm selected for use in the servocontroller is Pseudo-Derivative Feedback (PDF). A computer model and an operational laboratory model of a translational electrohydraulic position control system dynamically analogous to the structural, mechanical, and hydraulic components of the selected backhoe position control system were constructed. These models were exercised in simulation of the backhoe system equipped with an electronic servocontroller incorporating the PDF algorithm. The computer model was first validated in its baseline configuration by way of comparison with baseline laboratory model test results. "Baseline" refers to the complete system, but with proportional position feedback control in place of the PDF algorithm. An interesting aspect of this project is the two degree-of-freedom system constituted by the relatively compliant boom, coupling the hydraulic actuator with a fully loaded bucket. This is of particular interest since the bucket does not lend itself to position instrumentation, thereby precluding load position feedback. Data supporting the successful demonstration of computerized automation of optimal tuning is presented.

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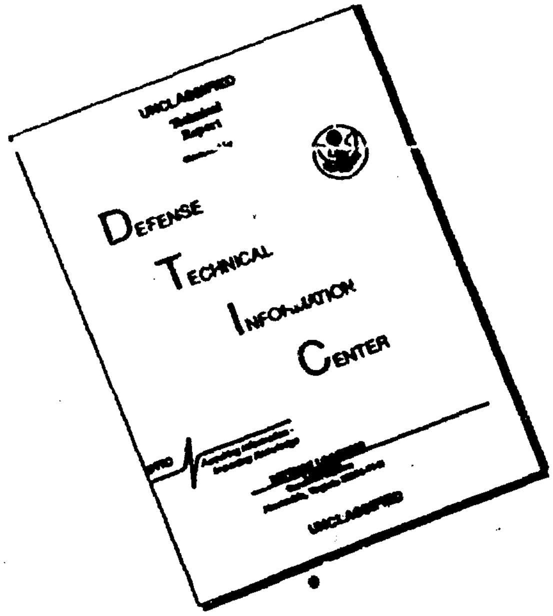

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METRIC CONVERSION FACTORS

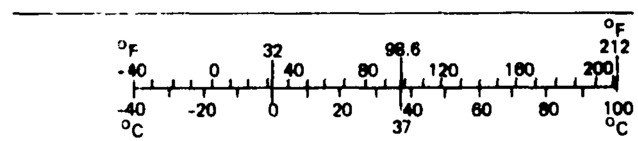
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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The objective of this work is the demonstration of feasibility of computerized optimal tuning of electronic motion controllers for mobile heavy equipment end-effectors. A backhoe yaw mode position control system was selected as the application for the purpose of proof of concept. The control algorithm selected for use in the servocontroller is Pseudo-Derivative Feedback (PDF). A computer model and an operational laboratory model of a translational electrohydraulic position control system dynamically analogous to the structural, mechanical, and hydraulic components of the selected backhoe position control system were constructed. These models were exercised in simulation of the backhoe system equipped with an electronic servocontroller incorporating the PDF algorithm. The computer model was first validated in its baseline configuration by way of comparison with baseline laboratory model test results. "Baseline" refers to the complete system, but with proportional position feedback control in place of the PDF algorithm. An interesting aspect of this project is the two degree-of-freedom system constituted by the relatively compliant boom, coupling the hydraulic actuator with a fully loaded bucket. This is of particular interest since the bucket does not lend itself to position instrumentation, thereby precluding load position feedback. Data supporting the successful demonstration of computerized automation of optimal tuning is presented.

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INTRODUCTION

Significant productivity and safety improvements in the operation of material handling and earth moving equipment are realizable through the use of modern electronic motion control of vehicle end-effectors. End-effectors include backhoe or excavator knuckle booms, material handling extendable booms, manipulators, or any of a myriad of attachments designed for use with these vehicles. The potential for greater productivity is based on increasing the operating speeds of end-effectors while limiting excessive overshoot and vibration through optimization of electronic controller design and performance. Similarly, there are opportunities for power and energy conservation through optimal tuning that focuses on these parameters. Concurrent with these benefits is the potential for improved reliability and life expectancy, based on reduced vibration of equipment.

BACKGROUND

Modern electronic motion controllers are essentially programmable microcomputers that utilize sophisticated control algorithms to achieve superior controlled system performance, i.e., faster response with minimum overshoot, or lower power consumption. These algorithms, in turn, typically utilize three or more gain settings in their component transfer functions which are somewhat arbitrary, and are therefore available for adjustment as part of any attempt to optimize performance. A mechanized/computerized method for determining an optimum set of algorithm gains (optimal tuning) is desired.

OBJECTIVE

It is the objective of this work to demonstrate the feasibility of a personal computer-based direct approach to the determination of an optimum set of values for electronic motion controller algorithm gains, i.e., optimal tuning, for applications involving mobile logistic heavy equipment such as backhoes, excavators, and material handlers. As a matter of interest, the equipment plant consists of a serial two degree-of-freedom system, i.e., two masses, in line and connected by a linear spring.

APPROACH

Optimization software (Ref 1), developed commercially for use in conjunction with control system analysis software (Ref 2), was installed on a Naval Civil Engineering Laboratory (NCEL) UNISYS type 386 personal computer. For comparison, the "Optimize" routine was also installed on an ISI owned and operated DEC workstation. A specific application in the area of mobile heavy equipment motion control was selected for study and evaluation of the feasibility of optimal tuning. The application selected is a backhoe boom position control system in the yaw mode, i.e., side-to-side swing. This operating mode constitutes a two degree-of-freedom system (two masses: (1) the hydraulic actuator, and (2) the effective boom and bucket/load, connected by a structural compliance, that of the boom in bending about a vertical axis). This application was selected because it represents one of the more difficult control problems in heavy equipment. A schematic of this single-axis system is shown in Figure 1. It is noted that the higher performance suggested by the use of a servocontroller requires that the conventional hydraulic valves be replaced with electrohydraulic servovalves (or proportional valves).

It was intended that the integrity of the computer model be validated by laboratory model simulation. Hence, a laboratory model was constructed for this purpose. A sketch showing the arrangement of the load assembly of the laboratory model test setup is shown in Figure 2. Full view and close-up view photographs of the laboratory model test setup are shown in Figures 3 and 4, respectively. An electronic circuit diagram of the laboratory servocontroller is shown in Figure 5; a photograph of the controller is shown in Figure 6. Laboratory model components are described in Appendix A. The laboratory model consisted of a translational electrohydraulic position control system that was dynamically analogous to the structural, mechanical, and hydraulic components of the selected backhoe boom position control system. Simplification to the translational mode was deemed expedient and adequate for proof of concept. These models (laboratory and computer) are based on the selection of a particular control algorithm for the purpose of demonstration and evaluation. The selected algorithm, Pseudo-Derivative Feedback (PDF), utilizes only three adjustable gains as compared to other more complex algorithms utilizing five or more gains. The PDF algorithm was therefore selected because of the relatively simpler problem it presents for optimization. No allowance is intended here for variability in the environment or load. The PDF algorithm is a concept of Richard M. Phelan, Professor of Mechanical Engineering, Cornell University (Ref 3). Figure 7 is a transfer function block diagram of the PDF algorithm integrated into a position control system.

A "Baseline" computer model, simulating the system, with proportional position feedback control in place of the PDF algorithm, was developed as well as a model with the PDF algorithm, hereinafter referred to as the "PDF" computer model. A comprehensive (expanded) computer model of the baseline system is depicted in block diagram form in Figure 8. A transfer function block diagram of the "Baseline" computer model, in "super block" form, is illustrated in Figure 9. The purpose served by the baseline model is to provide for model validation through laboratory model simulation runs prior to incorporating the added complexity of the PDF algorithm. A transfer function block diagram of the "PDF" computer model of this system is illustrated in Figure 10. Signal nomenclature for these block diagrams is given in Table 1. Gains and/or constants for the block diagrams are given in Table 2.

MODEL DEVELOPMENT

The baseline system parameters were selected to produce, approximately, the same fundamental natural frequency as the backhoe boom position control system in the yaw mode, and to assure system operation at roughly the same power level. Inasmuch as model validation was accomplished in two distinct parts ((1) the servovalve, and (2) the complete baseline system), model development is presented here in essentially the same order.

Servovalve Computer Validation Model

The servovalve computer validation model, which is different from the servovalve computer model, is illustrated in Figure 11. It is different in that standard servovalve calibration/test procedure requires that supply pressure be set at 1,000 psi. Since system operating pressure is 2,800 psi, this difference is reflected in these two computer models for this component. Also, load pressure for the validation model is taken as a constant, 80 psi, whereas it is a (variable) function of system operation in the case of the baseline model. The computer recognizes this model as "SVALVE." Servovalve flow constant calculations are shown in Appendix B.

Spring

The coupling spring, K_s, serves to connect the hydraulic actuator to the load. Its analog in the backhoe boom position control system is the structural compliance of the boom in pure bending about the vertical axis through its centroid. The spring used for the laboratory model was tested prior to use, and the calibration data are presented in Figure 12.

Baseline System

Closed-Loop Computer Model. The baseline system closed-loop computer model is shown in Figure 9. The forward path gain, P_k, is adjusted in the servocontroller in the lab model. The rest of the plant is represented by the "Translational Model" super block. Position feedback was taken as emanating from the hydraulic actuator (X_p) rather than from the load (X_l) since, in a real system, the load (bucket) does not lend itself to position instrumentation. The input gain, GC for "gain control," was set at "2" to compensate for the position transducer gain of "2" in the feedback loop. The purpose of "GC" is to normalize the output such that the static gain of the closed-loop system is unity. The "X_l Initial Condition" is provided to compensate for the static displacement of the load, in a vertical system orientation, due to its spring suspension.

Open-Loop Computer Model. The baseline system open-loop computer model is shown in Figure 13. (The gain, GC and the "X_l Initial Condition" do not enter into the open-loop system analysis.) The purpose of open-loop computer model construction was to perform stability analysis using root-locus methods. (The control system analysis program, MATRIX$_X$/PC, requires input of the open loop system for root-locus analysis.) The root locus for the baseline system computer model is shown in Figures 14 and 15; Figure 14 shows a comprehensive window while Figure 15 shows a close-up window of the dominant roots. The stability margin is determined to be 20 decibels (dB).

Expanded Super Blocks, Baseline System Computer Model. The "Translational Model" super block representing the plant in Figure 9 is shown expanded into its three component super blocks in Figure 16. Similarly, these three super blocks are expanded into their respective models in Figures 17, 18, and 19. Details of the modelling of the process mechanics are not discussed here since model development is not the primary objective of this work; it is left to the reader to correlate the model constructions at the component level with the operating characteristics of electrohydraulic position control systems.

PDF System

Closed-Loop Computer Model. The PDF system closed-loop computer model is shown in Figure 10. The PDF algorithm is contained in block numbers 97, 93, 11, 12, and 20; these correspond to the parameters K_i, K_r, K_1, K_2 and K_{fb}, respectively. To avoid noise generation in the simulation, the rate term in the PDF algorithm was implemented by tapping into the piston velocity signal, rather than by providing a differentiator.

Open-Loop Computer Model. The PDF system open-loop computer model is shown in Figure 20. Creation of the open-loop model required the conversion of the closed-loop model to a form characterized by a single feedback loop. Since the closed-loop model is a dual-loop feedback system, conversion to a single-loop system required consolidation of the two summing junctions into a single junction, for the outer loop only, using standard block diagram manipulation. As in the case of the baseline system, the purpose of open-loop computer model construction was to confirm system stability using root-locus methods. A root locus (Figures 21 and 22) indicated the need for a gain correction factor in the feedback loop of 0.02 in order to provide stable response with a damping ratio of 0.34 for the closed-loop system. After making this gain correction, it was found through a step response solution that the system was overdamped with an excessive settling time (0.7 second), so a gain correction factor of 3 was arbitrarily added to the outer feedback (proportional) loop to correct the problem.

Expanded Super Blocks, PDF System Computer Model. The "Translational Model" super block (Figure 23) is identical to that for the baseline system with the exception that there are two additional external outputs that are required for operation of the PDF algorithm. These are "piston acceleration" and "piston velocity." This difference also applies to one of its constituent super blocks, "Actuator-Load." The remaining two constituent super blocks, "Servovalve" and "Flow Continuity" are completely identical to their baseline system counterparts. It is noted that in the expanded Actuator-Load block diagram (Figure 24), the two additional external output terminals are shown superimposed on signal paths as terminal numbers 4 and 5 (software problem).

BASELINE MODEL VALIDATION

Physical characterization of the "Baseline" laboratory model was provided to NCEL by its test contractor (Ref 4). That characterization is summarized in terms of the nomenclature of Figure 8, and is presented in Table 1[1]. Attention is called to some major differences between the data furnished by the test contractor and that used for the computer model development. The two most important differences are discussed here. First, the leakage coefficient, K_{le}, was determined analytically to be considerably different from the value furnished. An analysis of internal actuator leakage is shown in Appendix C where K_{le} is shown to be equal to 0.010 in.^{3}/sec per psi. Use of this value for K_{le} was necessary in order to achieve reasonable agreement with laboratory model tests for purposes of model validation. Secondly, a viscous damper for the load was used in the computer model, while the test contractor actually used a

[1]Values of critical parameters also appear on system performance plots.

coulomb damper. It is believed that this shortcoming in computer modelling is responsible for the primary lack of agreement with laboratory model tests for purposes of system model validation.

The first step in model validation was validation of the servovalve step and frequency response. The servovalve computer validation model was evaluated for step response, i.e., valve output flow versus time; the results are plotted in Figure 25, along with laboratory test results obtained for the actual servovalve response to a step command. As shown in the plot, there is only a very minute difference between the two response functions. A similar comparison was then made for frequency response, and the results are plotted in Figure 26 where it is shown that the response functions are essentially coincident.

Results of the complete baseline laboratory and computer model simulations, in terms of system response to a step command in load position, are plotted in Figure 27. These results are shown as overlays plotted to the same scale for purposes of comparison. The baseline computer model included those nonlinearities deemed significant to the simulation. These plots show reasonably good agreement between the laboratory and computer models. The first overshoots are quite close in magnitude and phase for the two models; however, there are discrepancies in the system damping and natural frequency. These were calculated from the results presented in Figure 27; the calculations of system damping and natural frequency are presented in Appendix D. The discrepancy in natural frequency, ω_n, is not excessive, e.g., for the laboratory model, ω_n is 5.22 cycles per second, while for the computer model it is 4.88 cycles per second, a difference of 6.22 percent. The discrepancy in system damping is considered more significant; for the laboratory model, ζ is 0.142, while for the computer model it is 0.240. These differences are attributed to the difference noted above in the load damper (viscous damping for the computer model versus coulomb damping for the laboratory model). This notion can be confirmed by observing the peak amplitude decay for the laboratory model response function, and noting that it departs only a small amount from a straight line; coulomb damping is characterized by linear decay. On the other hand, the peak amplitude decay for the computer model response function is essentially exponential, as it should be for a viscous damper. However, the focus of this work is on the mechanization of an optimization routine, regardless of system configuration or character. Therefore, it was decided to accept the baseline computer model as valid for the purposes of this study. Results of the baseline laboratory and computer model simulations, in terms of frequency response, are presented in Figure 28.

PDF COMPUTER MODEL OPTIMAL TUNING

Computer solutions to the optimal tuning problem are based on randomly selected nonoptimal (initial/original) parameter sets, where the parameters are the adjustable coefficients of the selected algorithm. In Figure 29, the step response for the PDF computer model with the indicated nonoptimal parameter set is shown along with the baseline system step response for comparison. Although the nonoptimal PDF system clearly outperforms the baseline system, an optimal PDF system can do even better. This is made clear in Figure 30, where baseline system performance is compared with the performance of a PDF system which was optimized through trial and error. As suggested earlier, the objective of this work was to demonstrate the automation of optimal tuning using a commercially available computer program. In the five cases for which results are described below, individual computer diaries were obtained for the solutions, and these are presented in Appendix E. In the five cases described below, the results varied considerably. This is attributed to: (1) the use of alternate computational algorithms, (2) the use of alternate computers, and (3) variation in the number of computer iterations specified.

Case Number 1

In Figure 31, the results of exercising the "Optimize" routine for ISI's Matrix_x/PC program on the nonoptimal PDF parameter set of Figure 29 is shown. In this illustration, the step response corresponding to each of several successive iterations of parameter sets is shown, while the values for each parameter set are listed in Table 3, along with the percent overshoot for each case. For clarity, the step responses for the initial (nonoptimal) and final (optimal) parameter sets are shown in Figure 32. Referring to Figure 33 (which shows the multiplicative robustness[2] margin plot for the optimally-tuned PDF computer model), it is apparent that the improvement in performance achieved by optimal tuning does not come without a penalty, i.e., the robustness margin is diminished. The solution for Case Number 1 was obtained using an ISI owned and operated DEC workstation and a variable step Kutta Merson computer algorithm; the computer was programmed for one major iteration, and the solution time was 3.48 minutes. For the sake of comparison, the "optimal" PDF parameter set represented in Figure 30 was also analyzed for multiplicative robustness margin. Here, the subject system was tuned by trial and error, without the benefit of the optimization software routine used for all the other solutions. It is readily seen from Figure 30 that this parameter set provides a nearly ideal response, hence nearly ideal tuning. However, in Figure 34, which is the multiplicative robustness margin plot for this manually tuned system, it is shown that at the critical frequency of 5.46 cps, the margin is -1.43 dB. This does not compare favorably with the margin of +1.41 dB for the computer-tuned system, for which the robustness margin is plotted in Figure 33. These margins are compared at the same frequency, which is the critical frequency for both systems. Incidentally, it should be noted that inputing the manually tuned system to the computer for further optimization does not offer a significant benefit. This is evident from the difference in the two plots of Figure 34, and comparing that difference (0.11 dB) with the difference in plots in Figure 33 (6.62 dB).

It is therefore important to consider and compare several local optimum solutions that can be provided by computer-based optimal tuning. As an example, since the PDF parameter, K_2, is the coefficient of a rate term in an inner feedback loop, then noise introduced by way of its implementation should be considered when comparing local optimum solutions. However, robustness margin is also a consideration.

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[2]This is a plot of stability margin as a function of frequency. The margin is normally expressed in decibels (dB), and is the sum of margin contributions from both gain and phase. "Robustness Margin" is used to indicate the maximum allowable contribution to attenuation and phase shift of a given system by an "Uncertainty" transfer function in order to maintain stability. In the case of a multiplicative robustness margin, the "Uncertainty" transfer function must be cascaded with the plant in the forward path of the system's block diagram. (Other types of robustness margin pertain to an "Uncertainty" transfer function in a different location such as inner or outer feedback loops, or external inputs at various locations.) In this way, the effects of variability or uncertainty in the plant, the load, or operating environment on stability margin can be determined.

Case Number 2

In Figure 35, the results of a new exercise of the "Optimize" routine for ISI's Matrix_X/PC program on the nonoptimal PDF parameter set of Figure 29 is shown. The difference between Case Number 1 and Case Number 2 is in the computer algorithm used for the solution; in this case, it was a fixed step Kutta Merson algorithm. Table 4 shows that when using this algorithm, although the convergence of successive solutions was less regular than for Case Number 1, the "optimal" (final) solution provided for a step response with slightly less overshoot. Also, the computer was programmed again for one major iteration, and the solution time was 5.37 minutes. Again, for clarity, the step responses for the initial (nonoptimal) and final (optimal) parameter sets are shown in Figure 36.

Case Number 3

Case Number 3 is identical to Case Number 2 with the exception that the computer was programmed for four major iterations, rather than one, in order to determine the sensitivity of the final solution to the number of major iterations. Step responses for the initial (nonoptimal) and final (optimal) parameter sets are shown in Figure 37. Table 5 shows that the change in solution with the additional iterations is insignificant. Solution time was 9.83 minutes.

Case Number 4

Case Number 4 involves a new arbitrarily selected initial (nonoptimal) parameter set, with the solution performed on an NCEL owned and operated UNISYS type 386 personal computer equipped with 20 MB of RAM. This computer had a coprocessor speed of 25 MHz, and a hard disk capacity of 105 MB. Figure 38 shows the step responses for successive iterations of parameter sets, while the values for each parameter set are listed in Table 6, along with the percent overshoot for each case. For clarity, the step responses for the initial (nonoptimal) and final parameter sets are shown in Figure 39. In this case, the final solution is not nearly as ideal as was obtained for the previous cases, with the final solution overshoot at 4.43 percent. The cause of termination of iterations at a parameter set so remote from optimal is not known. It is noted that the convergence for successive solutions is much more regular than is the case for the fixed step Kutta Merson computational algorithm. The variable step Kutta Merson algorithm was used here, and the routine was completed in three major iterations in 1 hour and 51 minutes.

Case Number 5

Case Number 5 is identical to Case Number 4 with the exception that the fixed step Kutta Merson computational algorithm was used. Figure 40 shows the step responses for successive iterations of parameter sets, while the values for each parameter set are listed in Table 7, along with the percent overshoot for each case. For clarity, the step responses for the initial (nonoptimal) and final parameter sets are shown in Figure 41. Again, the use of the fixed step Kutta Merson computational algorithm results in a solution convergence pattern that is quite irregular. Also, although the final solution provides a step response with a much lower overshoot (2.27 percent), its path is far from being acceptable due to the time it takes to reach and maintain final position. The two major iterations required 2 hours and 17 minutes.

RESULTS

Results for the computer "Optimize" solutions for Case 1 through Case 5 are presented in Table 8. The various performance parameters listed in this table lead up to ΔP_g, the percent improvement in overall operating cycle productivity from the baseline and from nonoptimal PDF models, respectively. Inasmuch as ΔP_g has been computed for the overall operating cycle rather than for only the yaw movements, it was necessary to derive this parameter as a function of the original and final (optimal) yaw motion duty cycle. This derivation is presented in Appendix F. Results reported for ΔP_g are necessarily based on an assumed value for the yaw motion duty cycle; this was taken as 40 percent for the purposes of this study. However, alternate values can be assumed, and the equations in Appendix F can be used to compute the sensitivity of ΔP_g to yaw motion duty cycle. Computer solution times are included in Table 8 since they vary by a ratio of nearly 40:1 over the five cases. Comments and observations on the computer solutions for the five cases, with respect to their regularity of solution convergence, solution time, number of solution iterations, and computer type, are given in Table 9.

CONCLUSIONS

1. The objective of the project effort reported herein was accomplished to the extent that the existence of numerous local optimums would allow. Computerized automation of optimal tuning of mobile heavy equipment motion controllers (using the PDF algorithm) has been successfully demonstrated.

2. As a result of the studies reported herein, productivity gains on the order of 34 percent for the overall backhoe operating cycle (optimal case) compared to the nonoptimal case, and 42 percent compared to the baseline (non-PDF) case, are achievable through optimal tuning.

3. The workstation computer solution time for solving the optimal tuning problem for an arbitrary nonoptimal parameter set was minimal, i.e., between 3.5 and 10.0 minutes, depending primarily on the computational algorithm used. The personal computer solution time was considerably longer, but not unreasonable, i.e., approximately 2 hours.

4. The variable step Kutta Merson computational algorithm appears to offer a convergence of solutions characterized by greater regularity, and requiring less solution time than the fixed step Kutta Merson algorithm.

5. Acceptance of an optimal tuning solution, or optimal parameter set should not be based solely on the compensated system's dynamic response. The results of this effort point to the "trade-off" between dynamic response and robustness. Typically, improved dynamic response translates to a loss in robustness.

6. The problem of two degree-of-freedom control systems represented by the selected application is less manageable than previously expected due to the lack of commercially available cost-effective instrumentation for bucket/load position feedback. The best that can be hoped for is actuator position feedback.

7. Load position pseudo-feedback can be provided by modelling the control system "plant," including the boom, end-effector carriage, and load within the servocontroller. Using the actuator position signal as input, a simulated load position feedback signal can be generated.

8. The results of this effort and the degree of tuning achieved for the different cases studied do not appear to be adversely affected by the lack of load position feedback.

9. The two degree-of-freedom system representing the actuator and "plant" in the selected application is also encountered in material handling equipment, i.e., extendable boom forklifts and cranes. The problem is the same, which is that excessive boom flexure or a nonrigid tension line allows uncontrolled and unacceptably large random motion of the load.

10. Although optimal tuning in either the design or installation phase of mobile heavy equipment motion control can be considered a precursor to real-time automatic optimal tuning, the implementation procedure would be completely different. Rather than model the system, actual system performance signals would be fed back to the servocontroller, while load/environment signals would also be generated and provided to the controller, thereby comprising a real-time adaptive control system.

REFERENCES

1. Integrated Systems, Incorporated (ISI). "OPTIMIZE Routine" computer program. Santa Clara, CA.

2. ___________. "MATRIX$_X$/PC" Ver 8.0 computer program. Santa Clara, CA.

3. Richard M. Phelan. Automatic Control Systems. Ithaca, NY, Cornell University Press, 1977.

4. Test Systems and Simulation, Incorporated. Madison Heights, MI.

Table 1
Laboratory Model Block Diagram Signals

e(s)	=	error signal, volts
F_a	=	actuator force, lb
F_{ex}	=	external load damping force, lb
F_p	=	net force on actuator/piston, lb
F_{pa}	=	piston/actuator damping force, lb
F_s	=	spring force, lb
F_{sd}	=	structural damping force, lb
F_l	=	net force on load, lb
i	=	current to servovalve, ma
K_{v22}	=	pilot stage multiplying factor, psi^{½}
K_{v32}	=	valve spool multiplying factor, psi^{½}
P_l	=	load pressure, psi
Q_f	=	flapper flow, in.^{3}/sec
Q_{ld}	=	net servovalve flow to load, in.^{3}/sec
Q_{le}	=	leakage flow past piston, in.^{3}/sec
Q_p	=	pumping flow, in.^{3}/sec
Q_v	=	net servovalve flow, in.^{3}/sec
R(s)	=	controller input signal
T_{fb}	=	feedback torque, in.-lb
T_m	=	servovalve motor torque, in.-lb
T_n	=	net torque on servovalve armature, in.-lb
X_f	=	flapper displacement, in.
X_p	=	piston displacement, in.
$\dot{X}_p$	=	piston velocity, in./sec
$\ddot{X}p$	=	piston acceleration, in./sec^2
X_l	=	load displacement, in.
$\dot{X}_l$	=	load velocity, in./sec
$\ddot{X}_l$	=	load acceleration, in./sec^2
X_v	=	valve spool displacement, in.

Table 2
Laboratory Model Block Diagram Gains

SYMBOL	DEFINITION	UNITS	VALUE
A_p	piston/actuator area	$in.^2$	2.40
A_v	valve spool pressure area	$in.^2$	0.0769
β	bulk modulus of hydraulic oil	psi	141,000
B_p	piston/actuator damping coefficient	lb-sec/in.	74
B_1	structural damping coefficient	lb-sec/in.	0.30
B_2	external load damping coefficient	lb-sec/in	34
K_f	flex tube spring rate in servovalve	in.-lb/in.	260.7
K_{le}	leakage constant	$in.^3$/sec-psi	0.010
K_s	coupling spring constant	lb/in.	2161
K_t	position feedback transducer gain	volts/in.	2
K_{tm}	torque motor gain	in.-lb/ma	0.0053
K_{val}	Vickers valve flow constant	---------	0.675
K_{v21}	pilot stage flapper sensitivity	$in.^3$/sec-$lb^{1/2}$	4.82
K_{v31}	valve spool sensitivity	$in.^3$/sec-$lb^{1/2}$	70.8
K_w	servovalve constant	in.-lb/in.	72.2
M_p	mass of actuator piston	lb-sec^2/in.	0.121
M_1	mass of load	lb-sec^2/in.	2.13
P_k	forward path gain	ma/volt	353.5
P_s	supply pressure	psi	2800
V_t	volume of oil under compression	$in.^3$	33.3
W_p	piston weight	lb	46.7
W_1	load weight	lb	822

Note: The term s in the block diagram represents the Laplacian operator and has units of sec^{-1}.

Table 3
PDF Parameter Sets for Case No. 1

ISI Variable Step Kutta Merson Solution
1 Major Iteration

Curve Number	K_1	K_i	K_2	Percent Overshoot
1	12.0000	1.0000	0.3000	12.00
2	12.0219	0.9511	0.1671	9.89
3	12.0432	0.8992	0.0264	7.60
4	12.0630	0.8462	-0.1171	5.20
5	12.2771	0.8640	-0.2456	3.23
6	12.5345	0.8681	-0.4483	0.99

NOTES:
1. In Tables 3 through 7, K_1 = proportional gain, K_i = integral gain, and K_2 = derivative gain.
2. Curve 1 is the original parameter set and curve 6 is the optimal set.

Table 4
PDF Parameter Sets for Case No. 2

ISI Fixed Step Kutta Merson Solution
1 Major Iteration

Curve Number	K_1	K_i	K_2	Percent Overshoot
1	12.0000	1.0000	0.3000	12.00
2	12.0254	0.8753	0.1960	9.35
3	12.0413	0.9345	0.0439	8.21
4	12.1318	0.9487	-0.6525	7.57
5	12.0913	0.9061	-0.2870	3.16
6	12.1325	0.8373	-0.4754	0.94

NOTE:
1. Curve 1 is the original parameter set and curve 6 is the optimal set.

Table 5
PDF Parameter Sets for Case No. 3

ISI Fixed Step Kutta Merson Solution
4 Major Iterations

Iteration Number	K_1	K_i	K_2	Percent Overshoot
Original Set	12.0000	1.0000	0.3000	12.00
1	12.5641	0.8699	-0.4456	0.99
2	12.5646	0.8699	-0.4460	0.99
3	12.5649	0.8698	-0.4462	0.99
4	12.5649	0.8698	-0.4462	0.99

<u>NOTE:</u>
1. The additional major iterations did not change the parameter sets very much.

Table 6
PDF Parameter Sets for Case No. 4

NCEL Variable Step Kutta Merson Solution

Curve Number	K_1	K_i	K_2	Percent Overshoot
1	8.0000	0.7500	0.2000	19.77
2	8.0331	0.6989	0.0491	16.99
3	8.0660	0.6422	-0.1178	13.59
4	8.0984	0.5786	-0.3044	9.35
5	8.1340	0.5131	-0.5035	4.43

<u>NOTE:</u>
1. Curve 1 is the original parameter set and curve 5 is the final set.

Table 7
PDF Parameter Sets for Case No. 5

NCEL Fixed Step Kutta Merson Solution

Curve Number	K_1	K_i	K_2	Percent Overshoot
1	8.0000	0.7500	0.2000	19.77
2	8.0408	0.5350	0.1058	14.83
3	8.0704	0.2765	0.0485	12.80
4	8.0306	0.5886	0.1293	2.27

NOTE:
1. Curve 1 is the original parameter set and curve 4 is the final set.

Table 8
Summary of Results, Optimal Tuning

Performance Parameter	Case Number				
	1	2	3	4	5
Computer Solution Time	00:03:28	00:05:22	00:09:50	01:50:48	02:17:34
Percent Overshoot	0.99	0.94	0.99	4.43	2.27
T_s, Nonoptimal PDF	0.41	0.41	0.41	0.71	0.702
T_s, Optimal PDF	0.15	0.15	0.15	0.42	0.726
ΔT_{dc} Baseline	-63.2%	-63.2%	-63.2%	-18.6%	13.7%
ΔT_{dc} Nonoptimal PDF	-51.0%	-51.0%	-51.0%	-29.3%	2.02%
ΔP_g Baseline	42.2%	42.2%	42.2%	12.4%	-9.15%
ΔP_g Nonoptimal PDF	34.0%	34.0%	34.0%	19.5%	-1.45%

NOTES:
1. The solution time is expressed in hours, minutes, and seconds.
2. T_s is the settling time and is taken to be the time needed to reach within 1.25% of the final postion.
3. ΔT_{dc}, baseline and ΔT_{dc}, nonoptimal PDF are the percent improvements in yaw duty cycle time from the baseline and nonoptimal PDF models, respectively.
4. ΔP_g, baseline and ΔP_g, nonoptimal PDF are the percent improvements in the overall productivity from the baseline and nonoptimal PDF models, respectively.
5. An initial yaw duty cycle of 40 percent was assumed for these calculations.

Table 9
Comments and Observations, Optimal Tuning

Case Number	Comments and Observations
1	ISI-VKM workstation solution, (WS), 1 major iteration, uniform convergence pattern, fastest solution time.
2	ISI-FKM WS, 1 major iteration, scattered convergence pattern.
3	ISI-FKM WS, 4 major iterations, no significant change in performance with additional iterations, solution time is almost doubled.
4	NCEL-VKM 386 personal computer (PC) solution, uniform convergence pattern.
5	NCEL-FKM 386 PC, scattered convergence pattern, optimal solution has no undershoot and is overdamped, the productivity and cycle time worsen slightly but this is offset by a lower overshoot.

NOTES:
1. ISI (Integrated Systems, Inc.) is a computer software company which was contracted to provide the optimization routines for the computer model.
2. VKM and FKM were the integration algorithms Variable Step Kutta Merson and Fixed Step Kutta Merson used in the computer solutions.

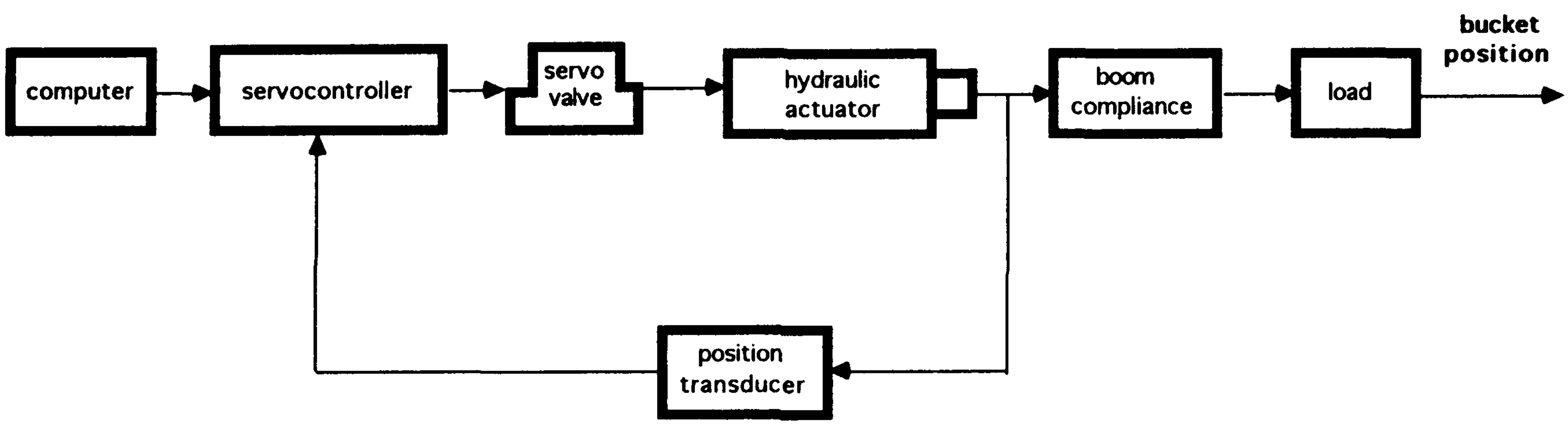


Figure 1
Laboratory model: system schematic block diagram.

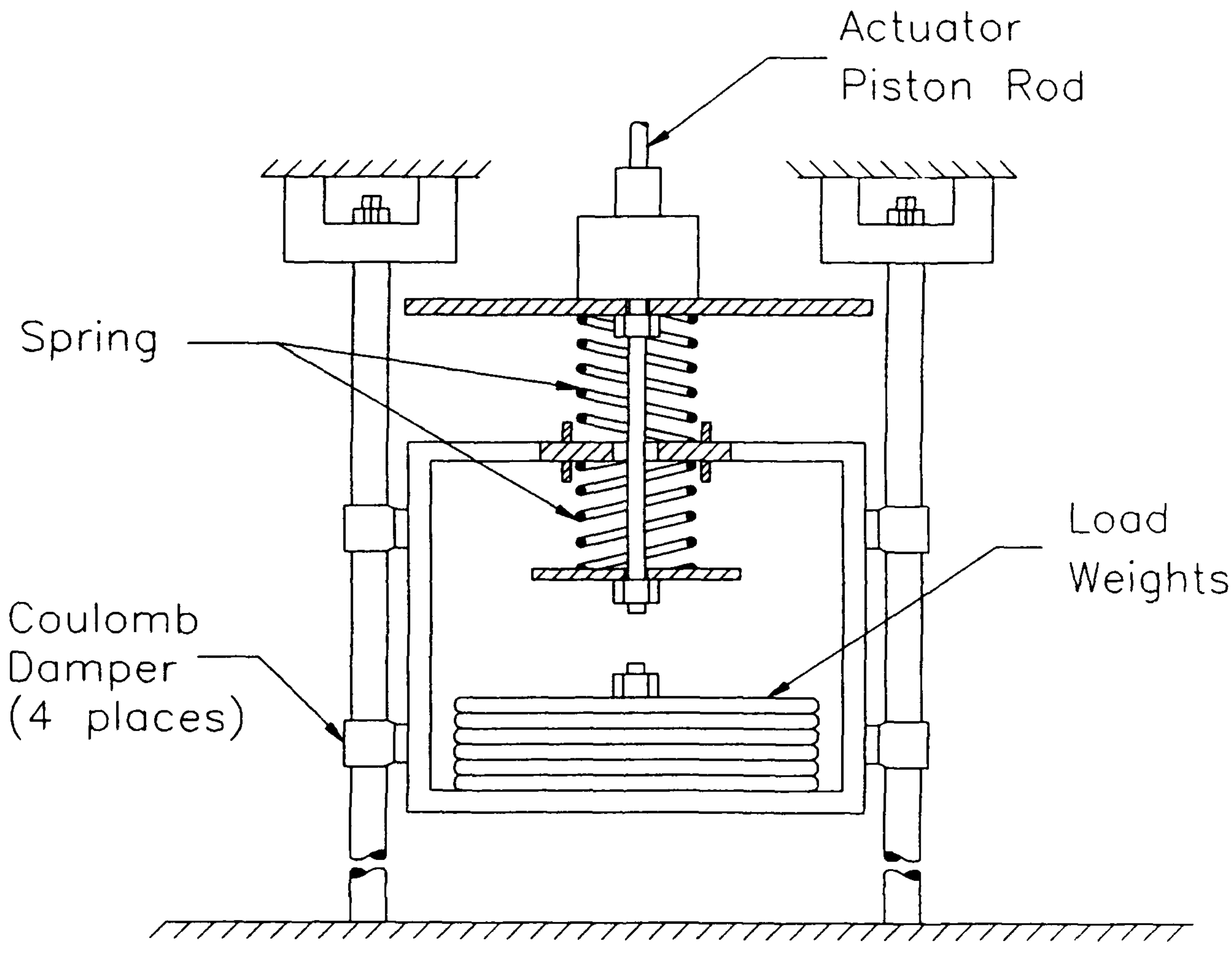


Figure 2
Sketch, laboratory model load assembly.

Figure 3
Photograph, laboratory model test rig.

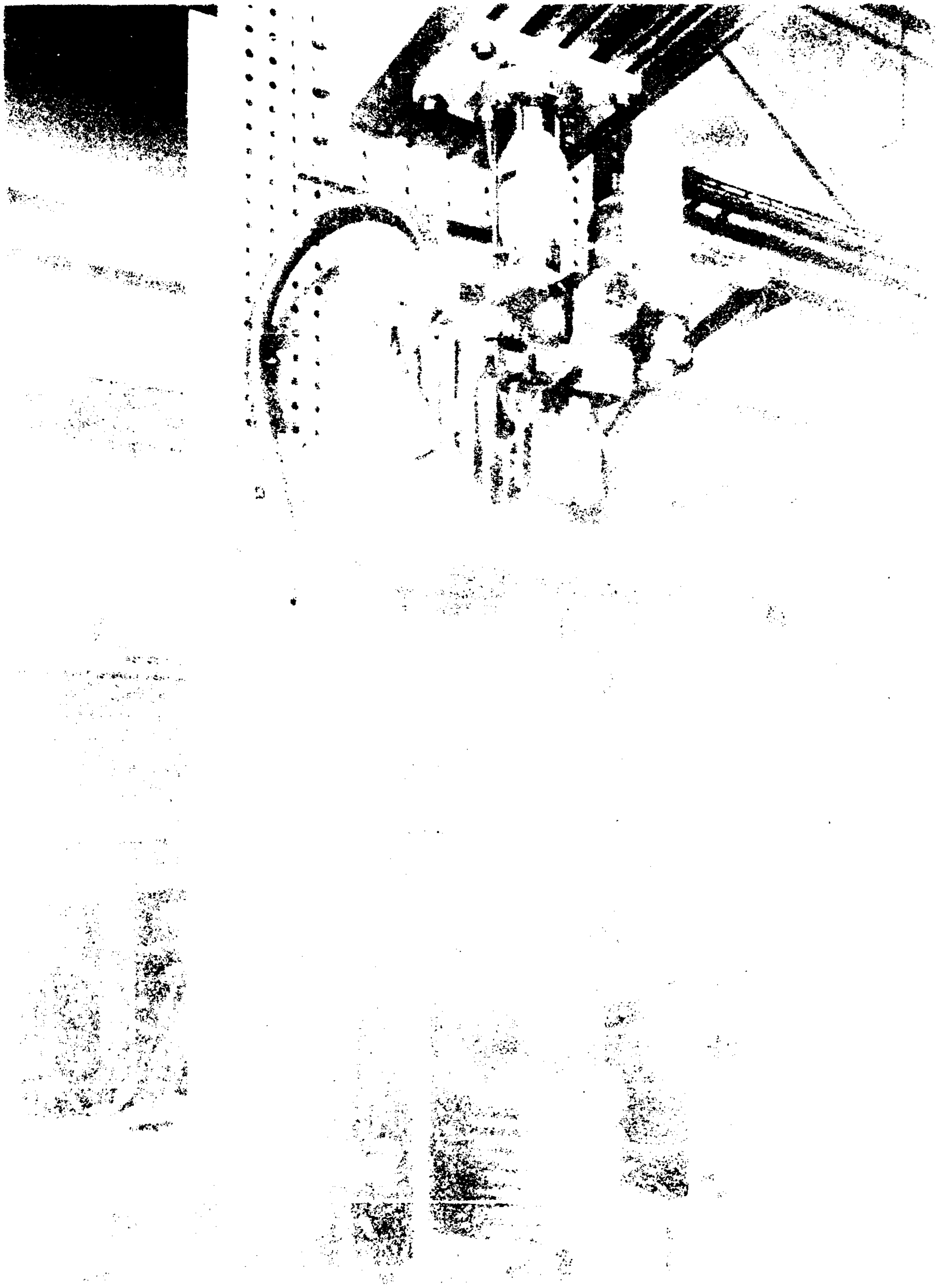


Figure 4
Photograph, laboratory model test rig load assembly

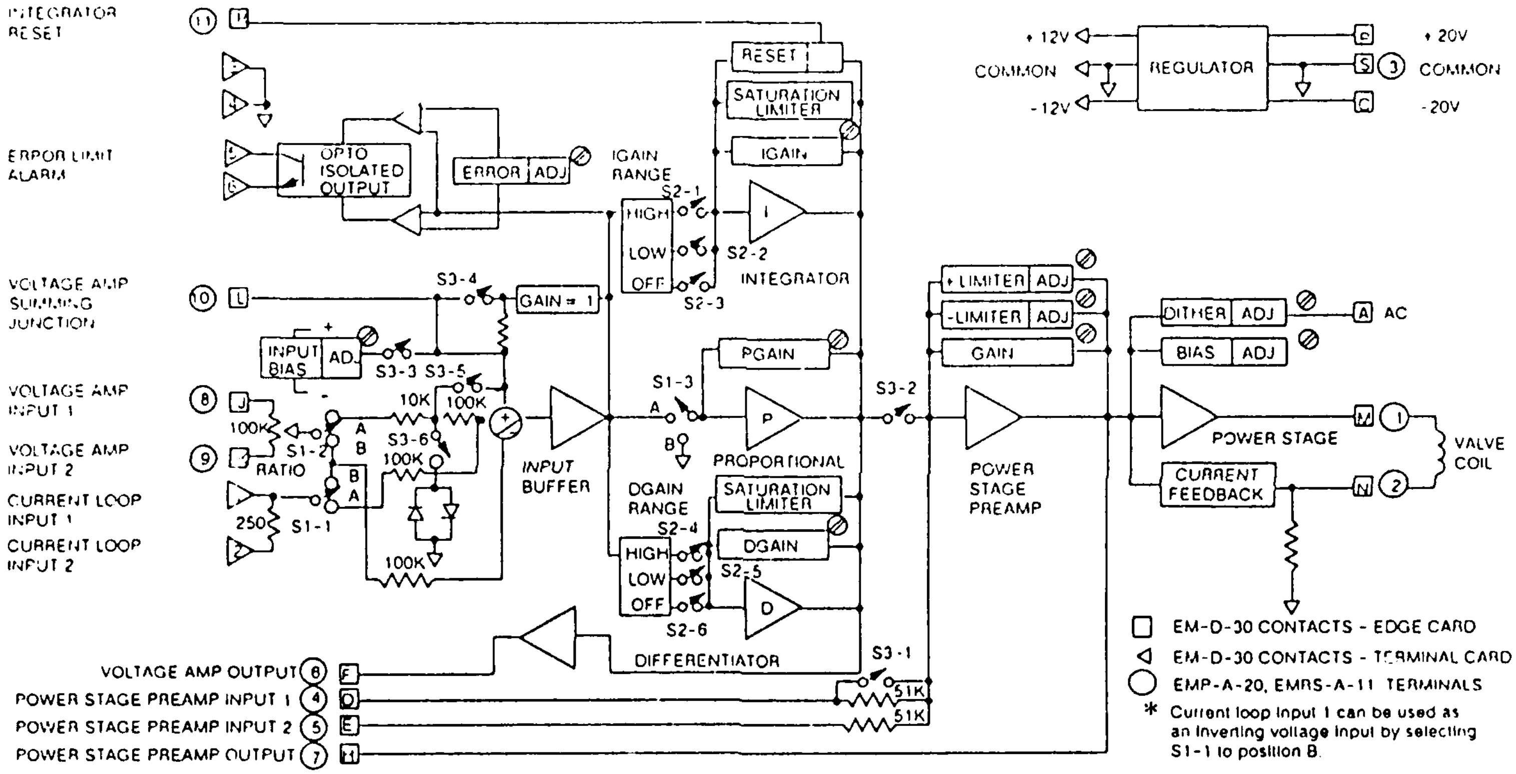


Figure 5
Electronic servocontroller circuit diagram.

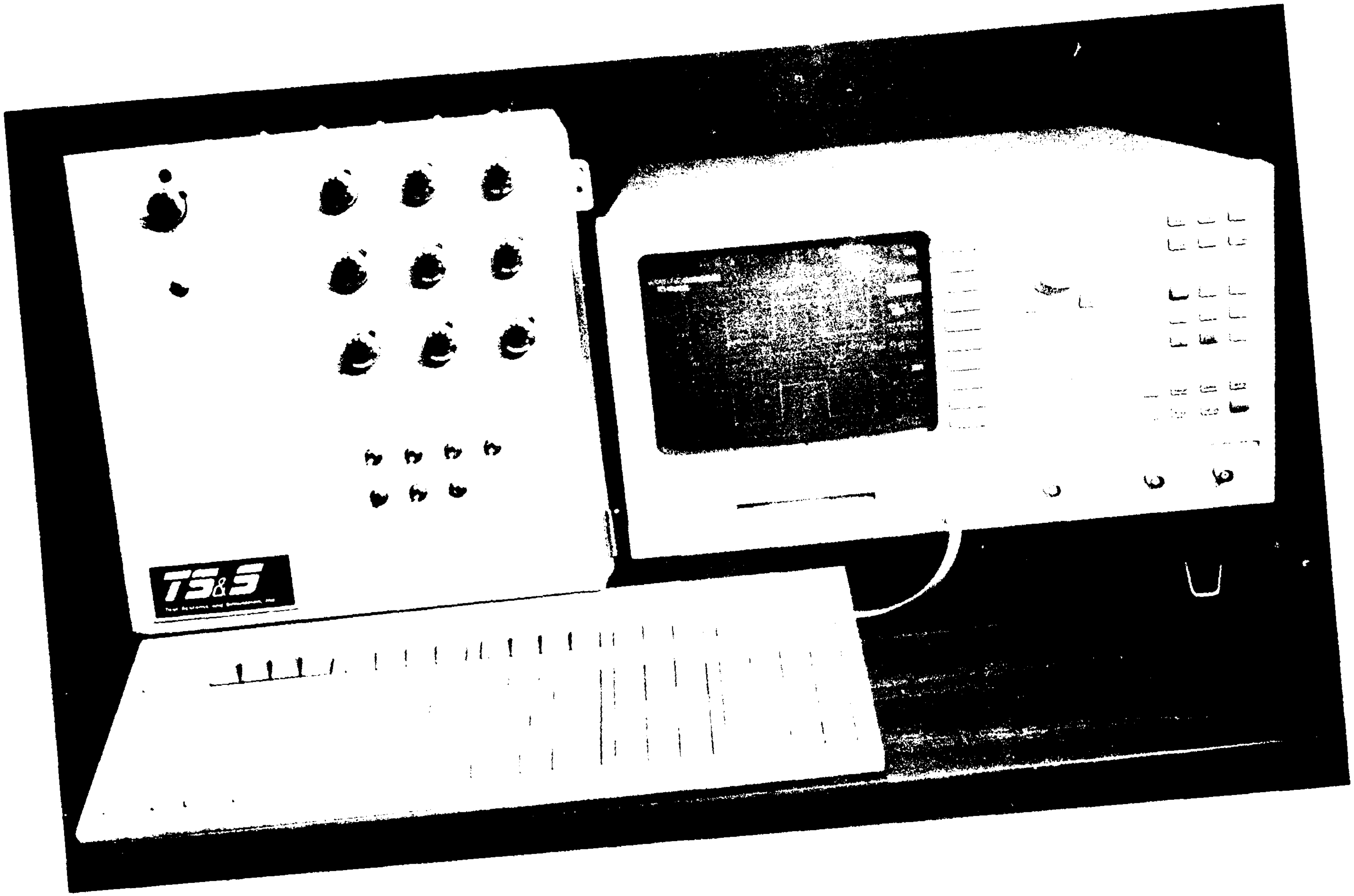
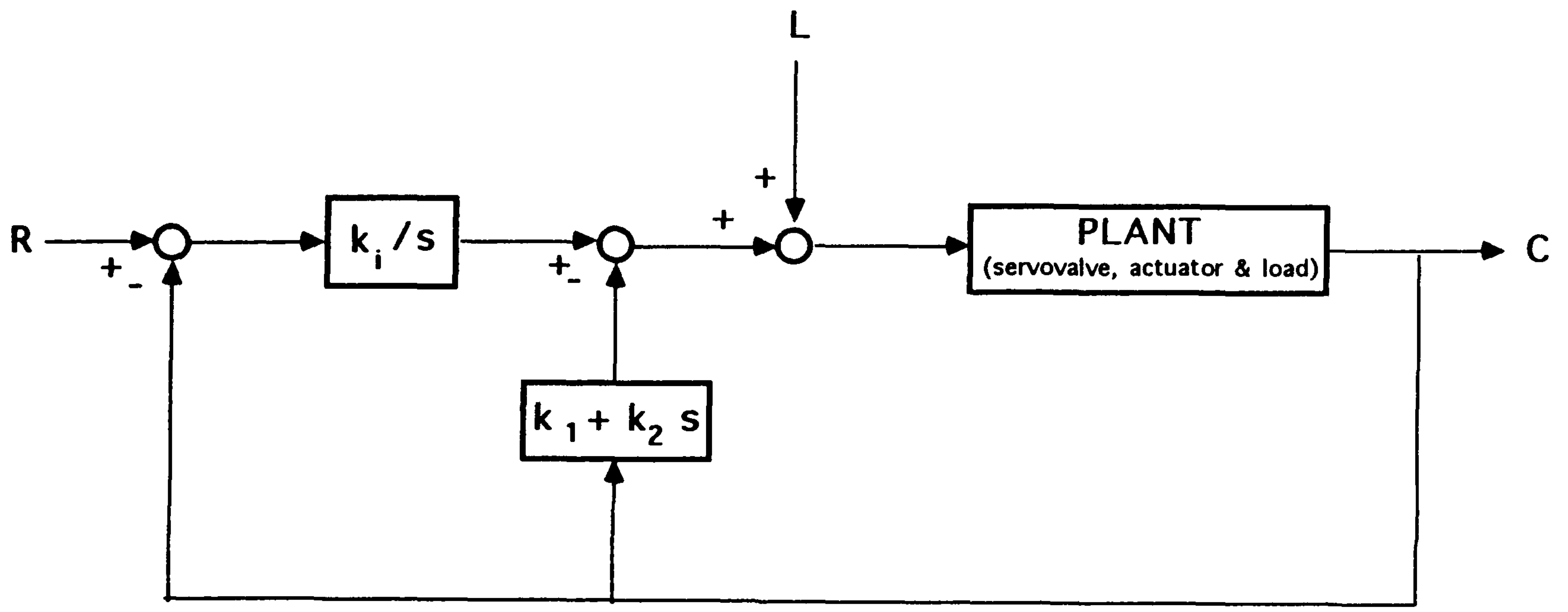


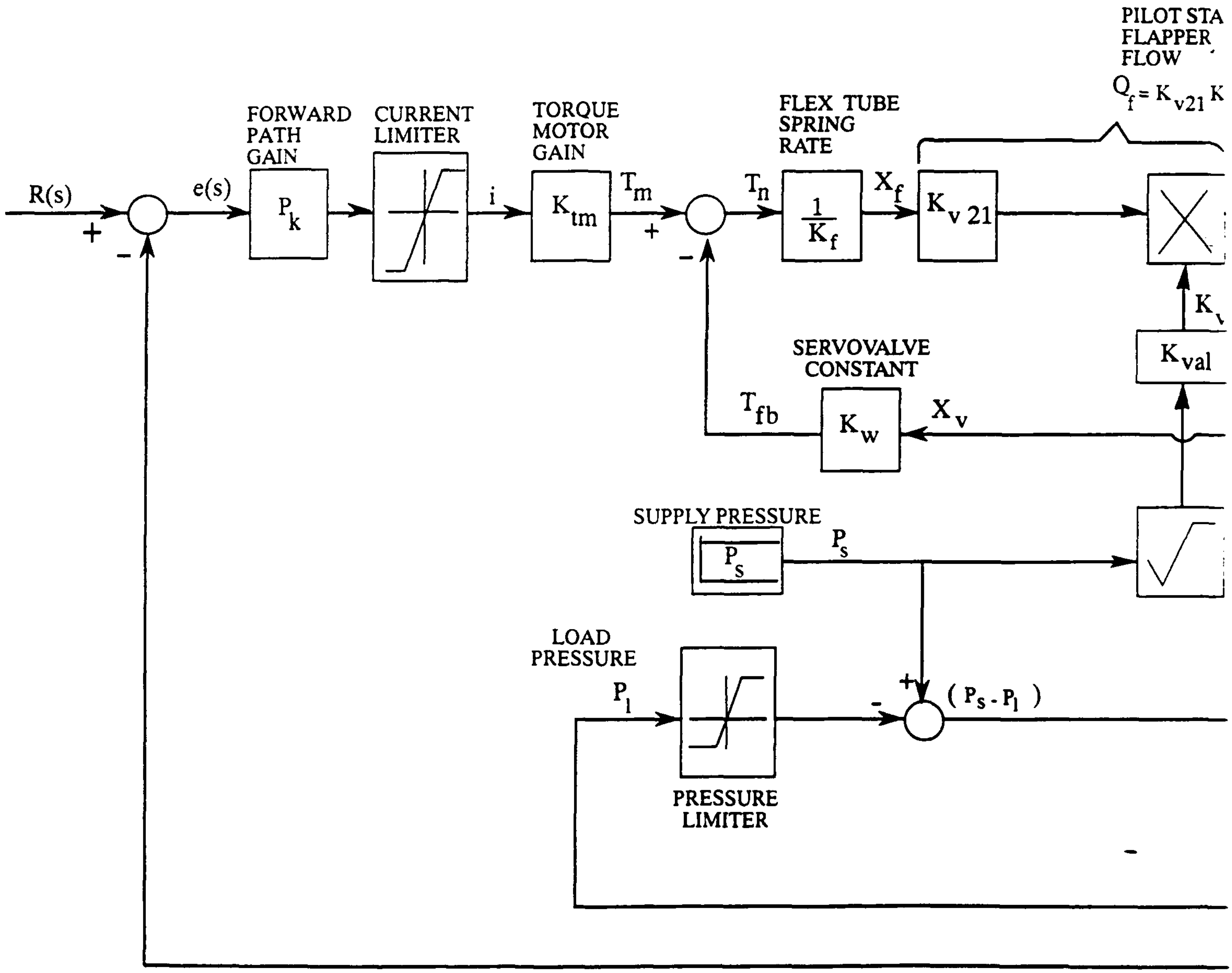
Figure 6
Photograph, electronic servocontroller (and digital signal analyzer).



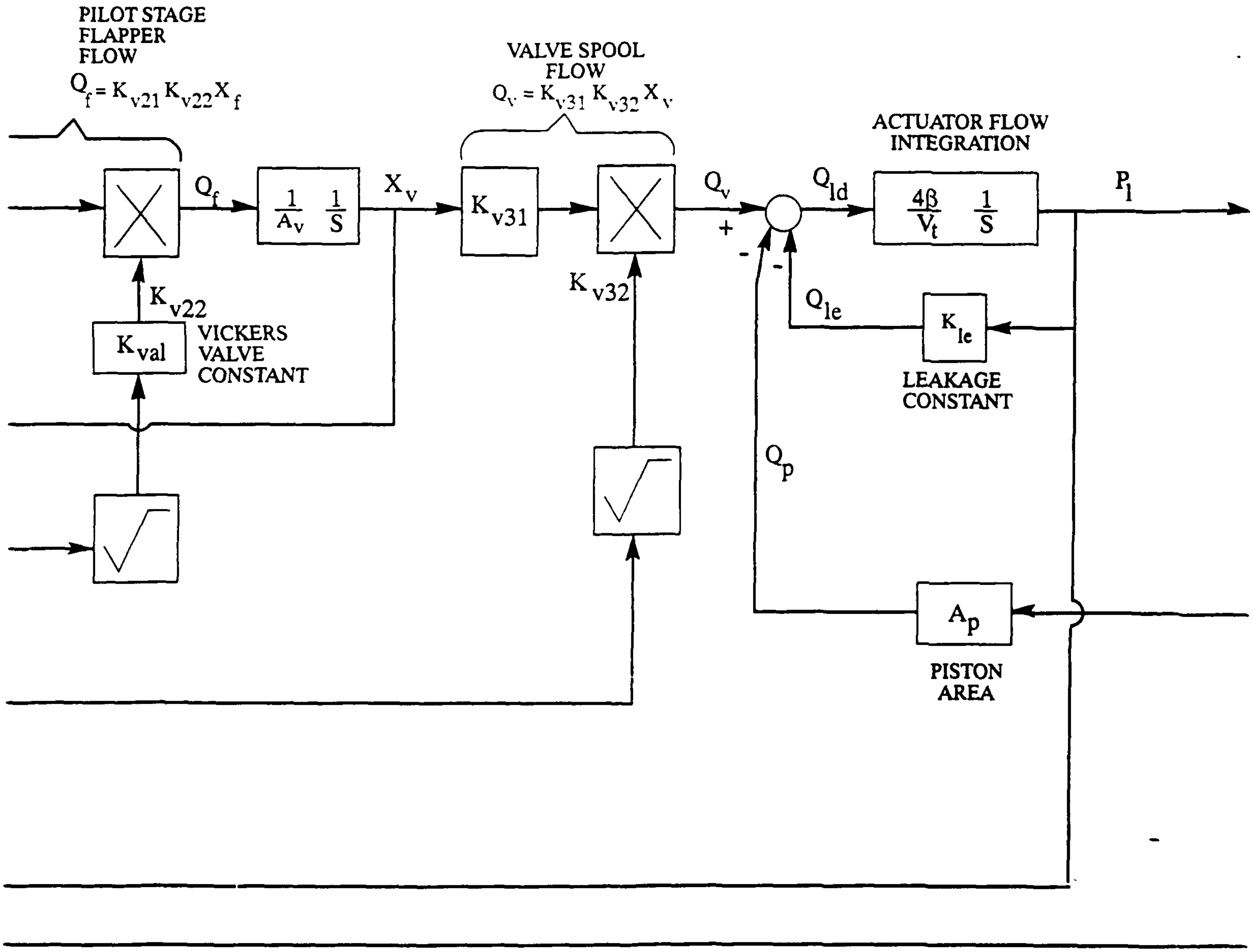
R=Reference (input) signal
C=Controlled (output) signal
L=Load disturbance

k_i = integral gain
k_1 = proportional gain
k_2 = derivative gain
s = Laplacian operator

Figure 7
Pseudo-derivative feedback control system.



PILOT STA
FLAPPER
FLOW
$Q_f = K_{v21} K$
FORWARD PATH GAIN
CURRENT LIMITER
TORQUE MOTOR GAIN
FLEX TUBE SPRING RATE
R(s)
e(s)
P_k
i
K_{tm}
T_m
T_n
$\frac{1}{K_f}$
X_f
$K_{v\,21}$
K_v
K_{val}
SERVOVALVE CONSTANT
T_{fb}
K_w
X_v
SUPPLY PRESSURE
P_s
LOAD PRESSURE
P_l
PRESSURE LIMITER
$(P_s - P_l)$



PILOT STAGE
FLAPPER
FLOW
$Q_f = K_{v21} K_{v22} X_f$
VALVE SPOOL
FLOW
$Q_v = K_{v31} K_{v32} X_v$
ACTUATOR FLOW
INTEGRATION
Q_f
$\frac{1}{A_v}$ $\frac{1}{S}$
X_v
K_{v31}
Q_v
Q_{ld}
$\frac{4\beta}{V_t}$ $\frac{1}{S}$
P_l
+
-
-
K_{v22}
K_{val}
VICKERS
VALVE
CONSTANT
K_{v32}
Q_{le}
K_{le}
LEAKAGE
CONSTANT
Q_p
A_p
PISTON
AREA

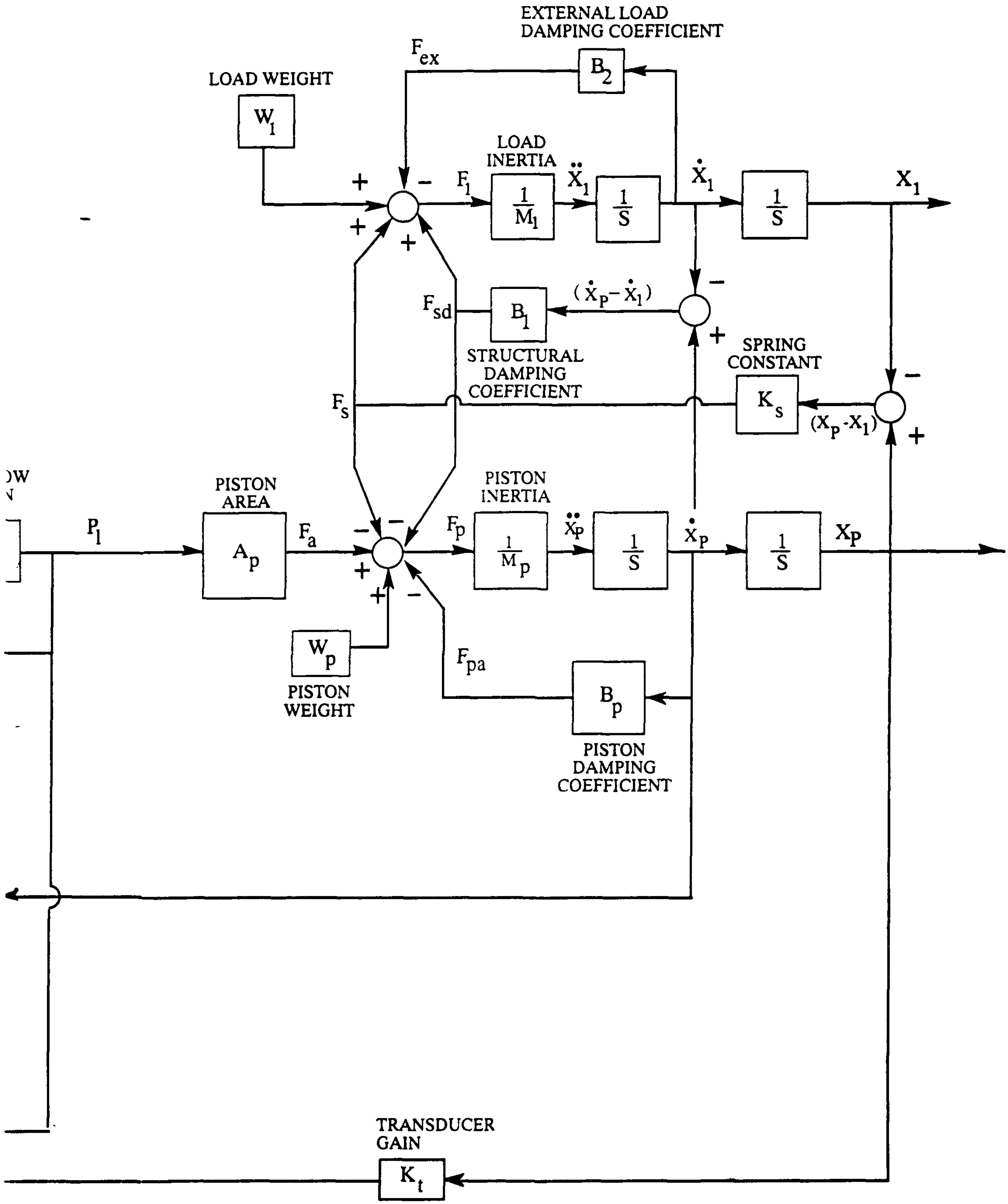


Figure 8
Expanded baseline system computer model.

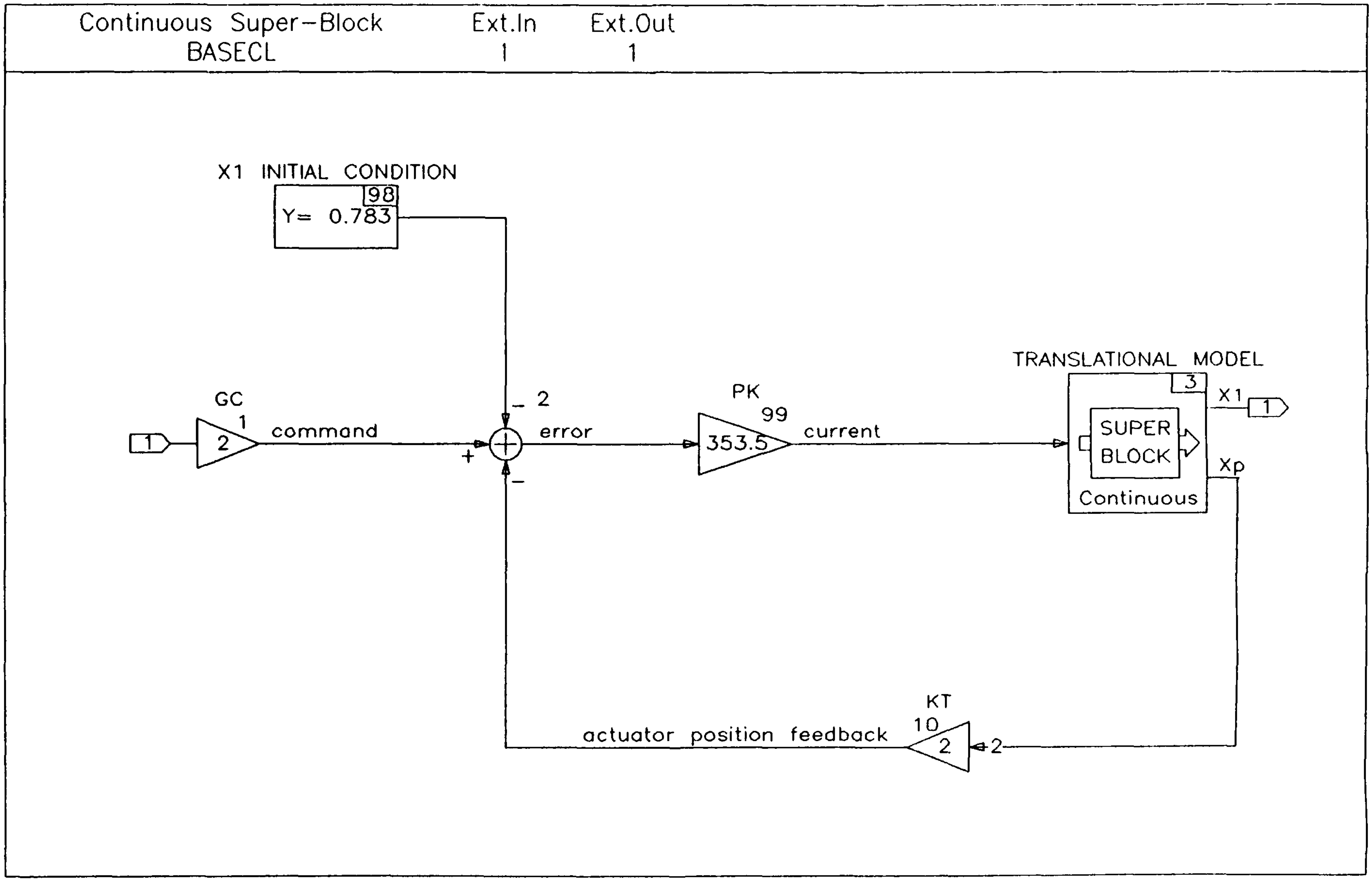


Figure 9
Baseline closed-loop system computer model.

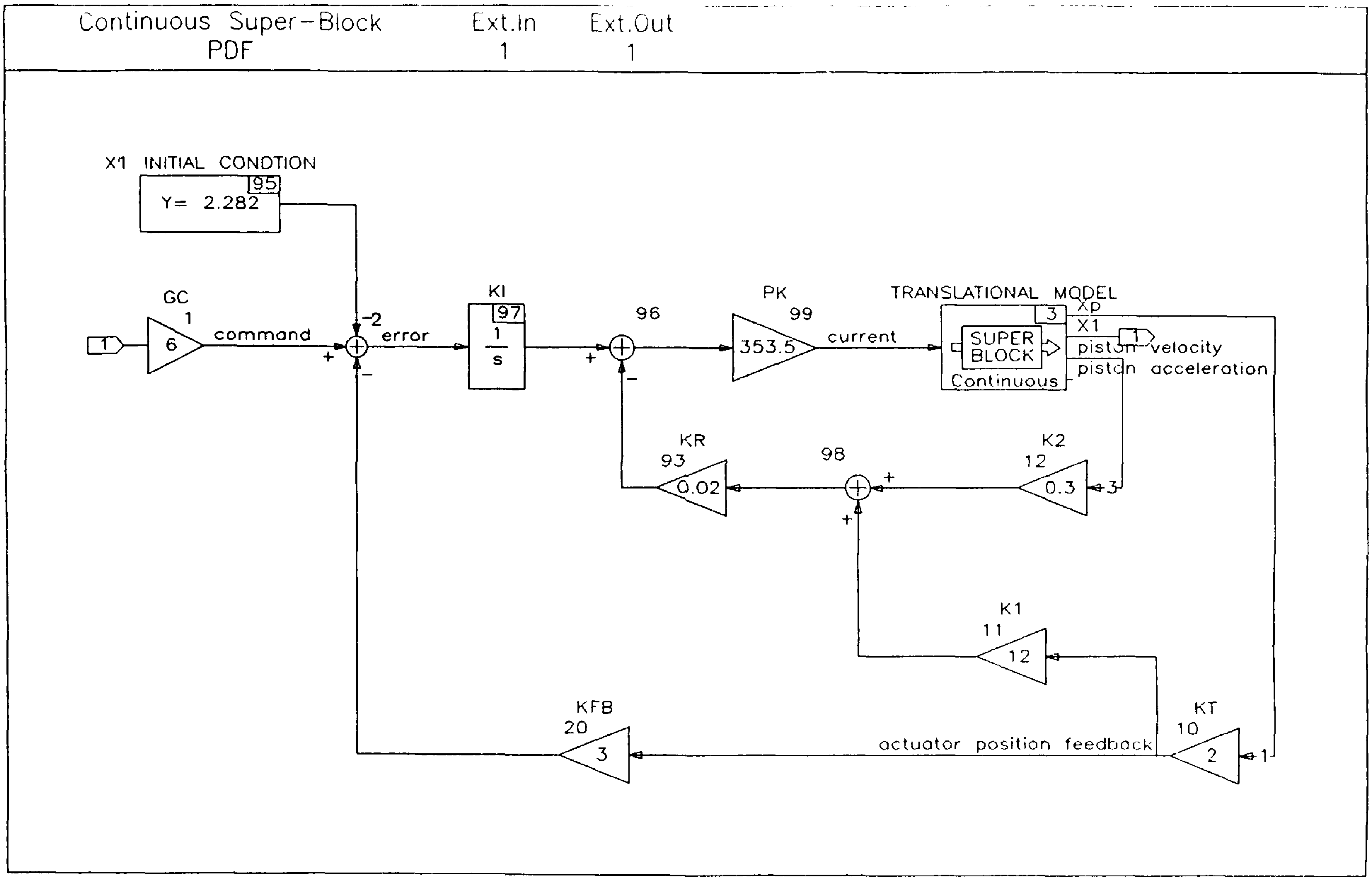


Figure 10
PDF closed-loop system computer model.

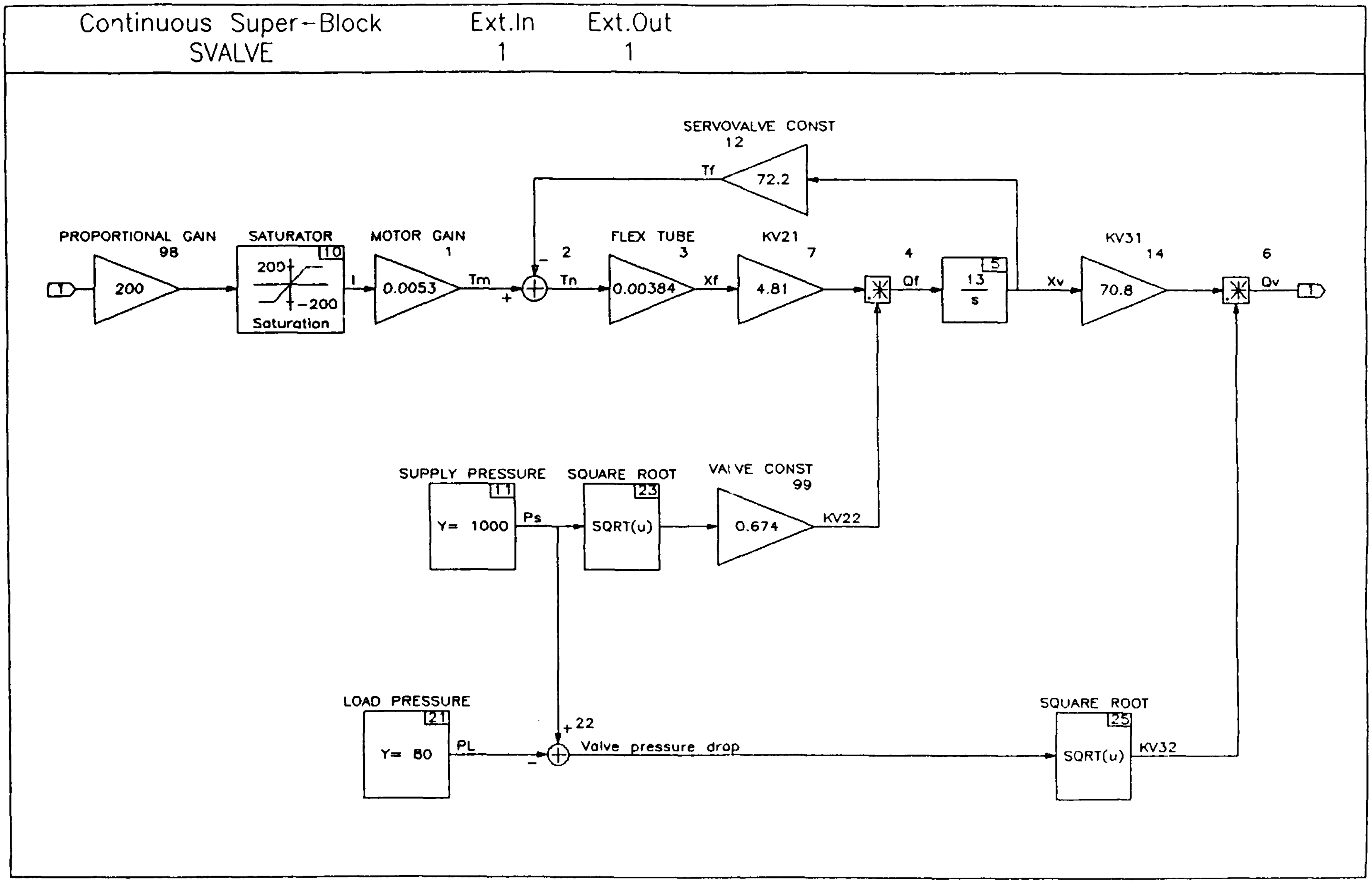


Figure 11
Servovalve computer validation model.

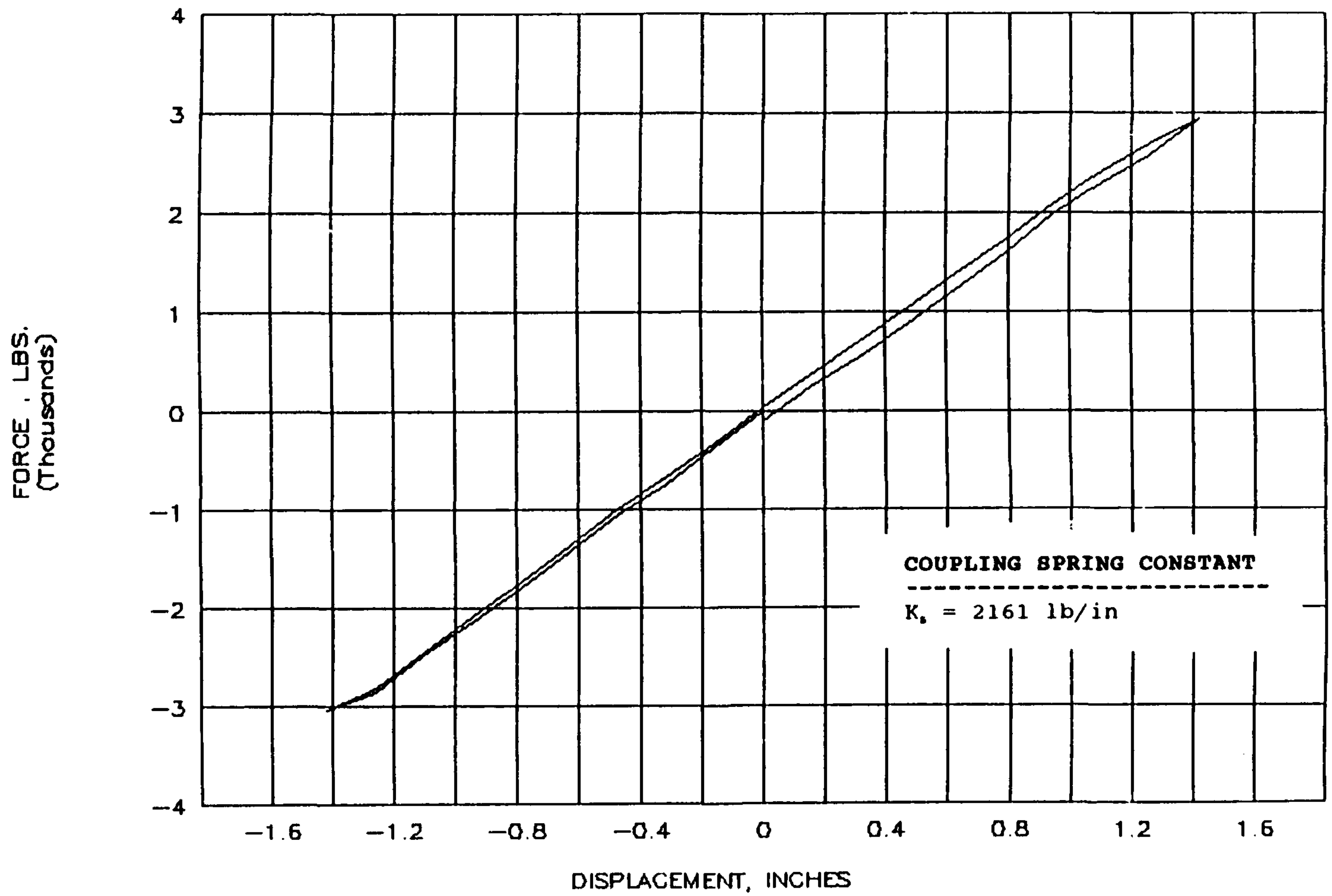


Figure 12
Coupling spring (K_s) calibration.

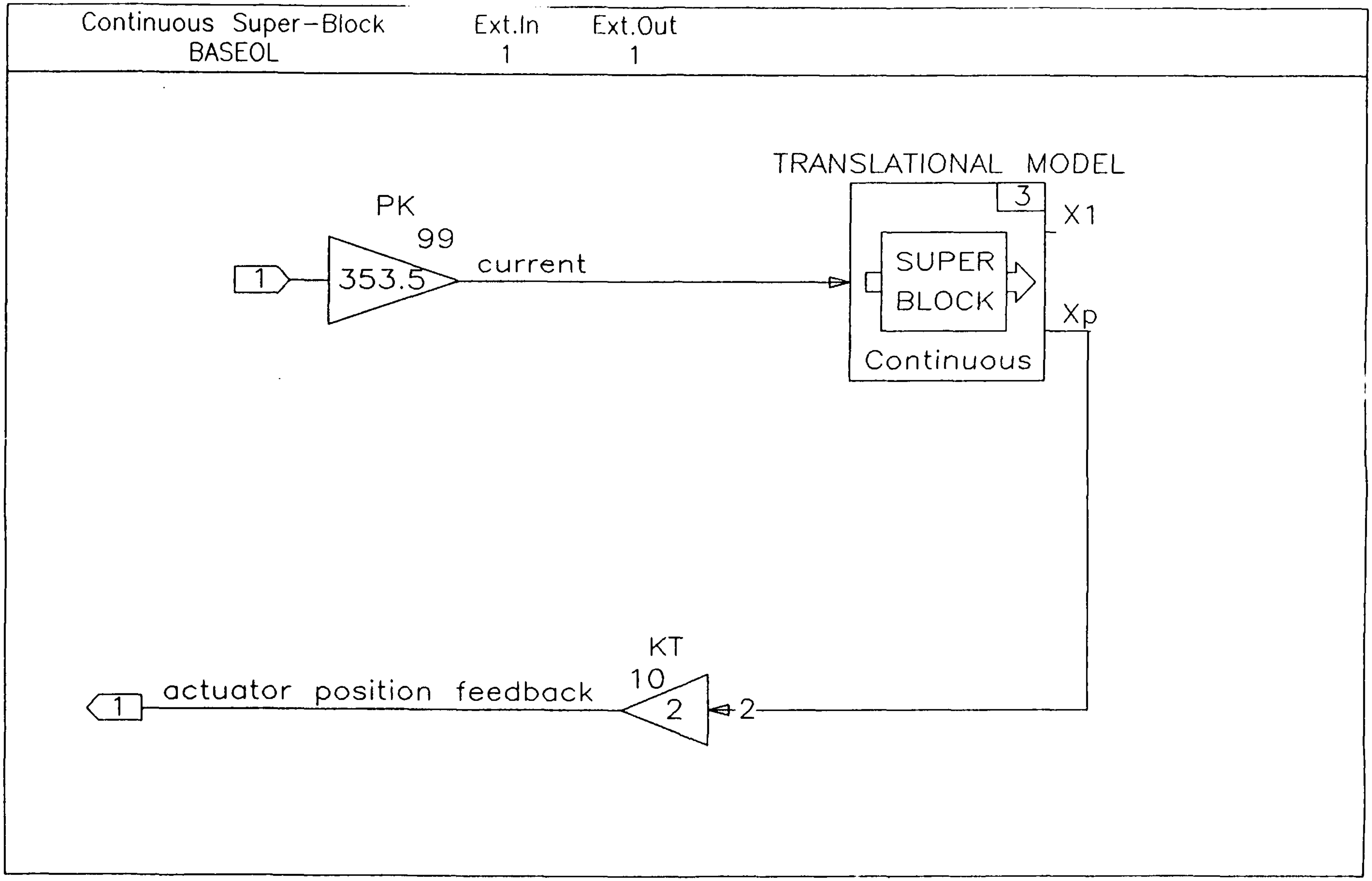


Figure 13
Baseline open-loop system computer model.

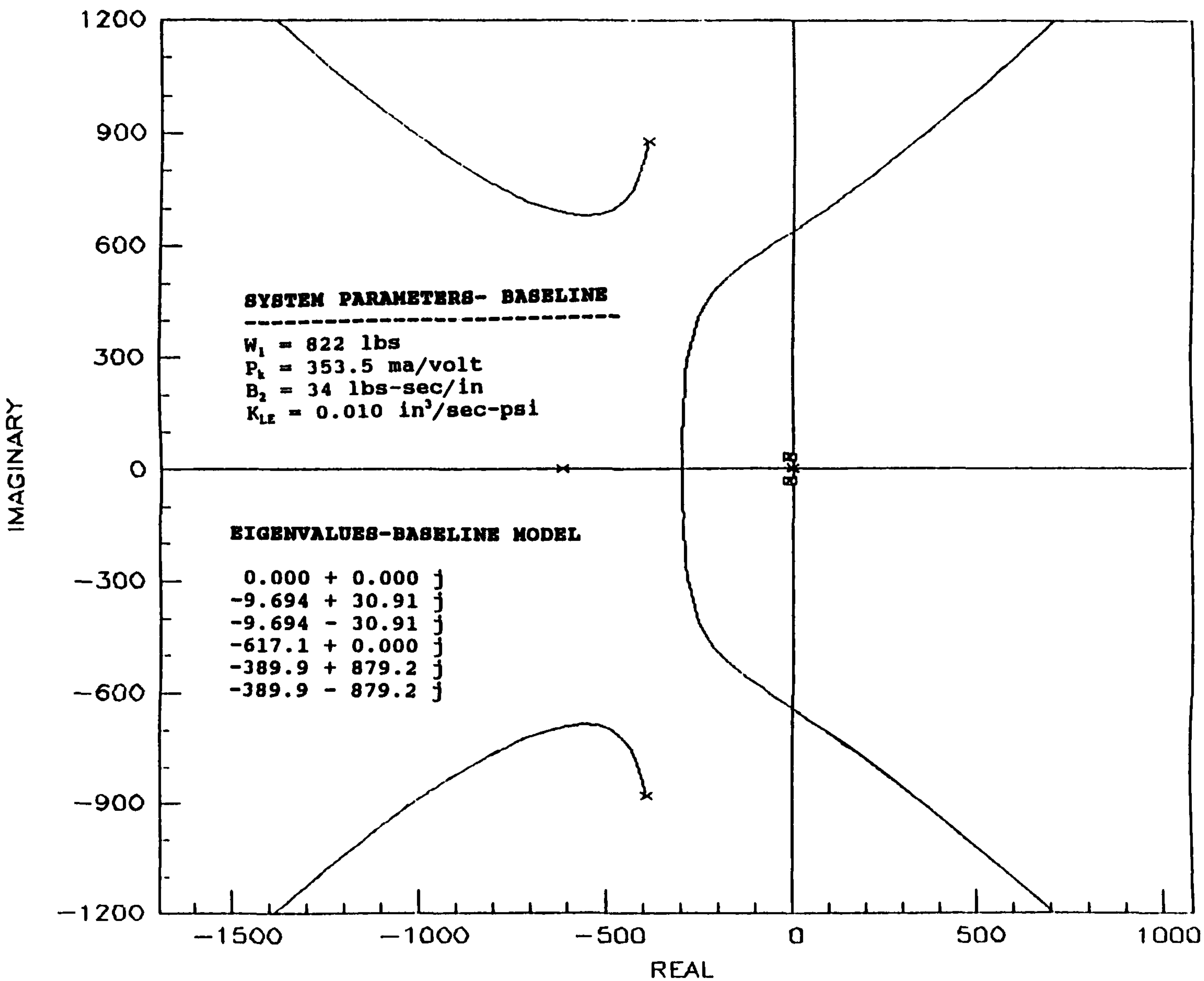


Figure 14
Baseline system computer model: root locus.

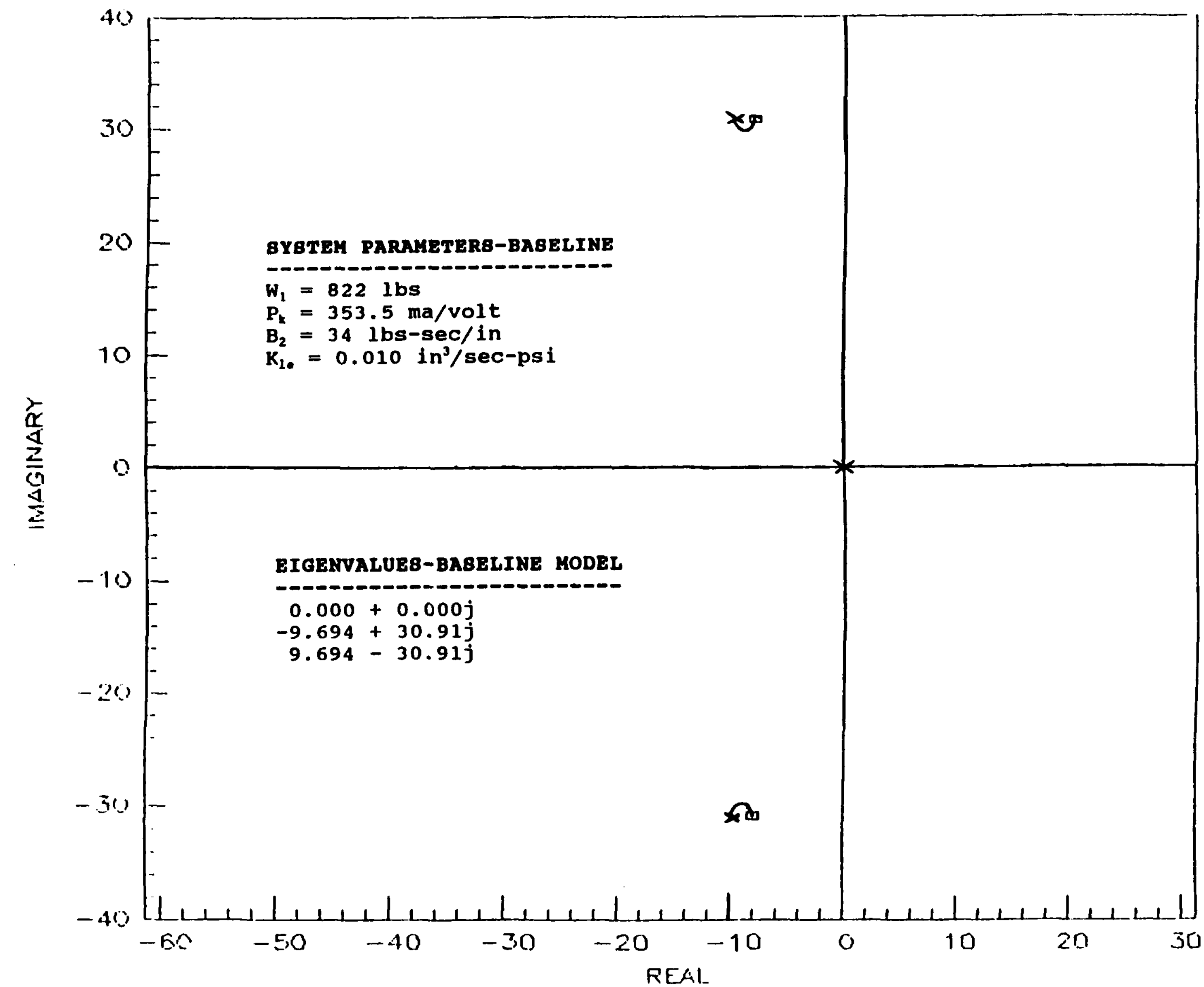


Figure 15
Baseline system computer model: root locus, dominant roots.

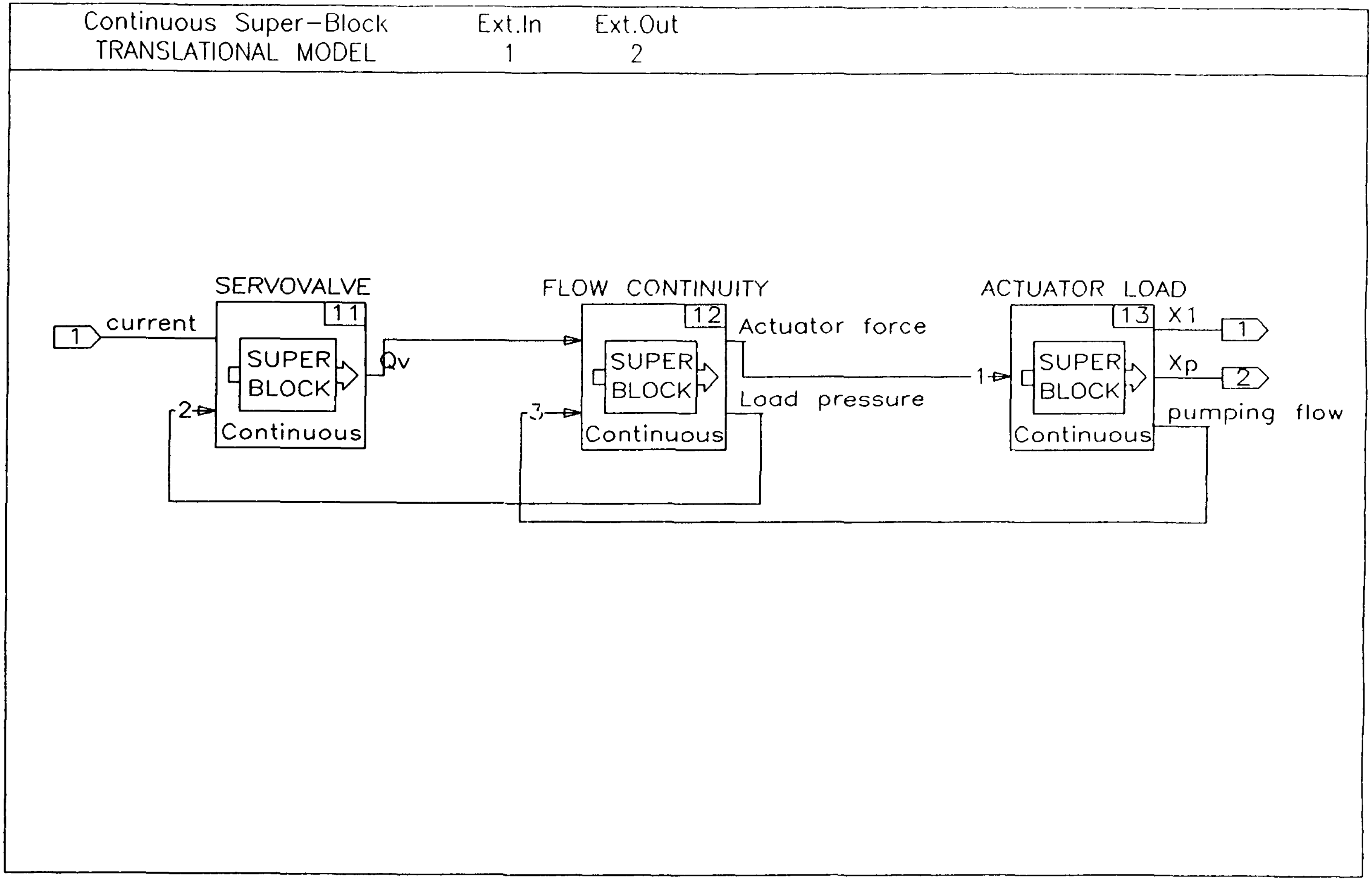


Figure 16
Baseline translational model: super-block expansion.

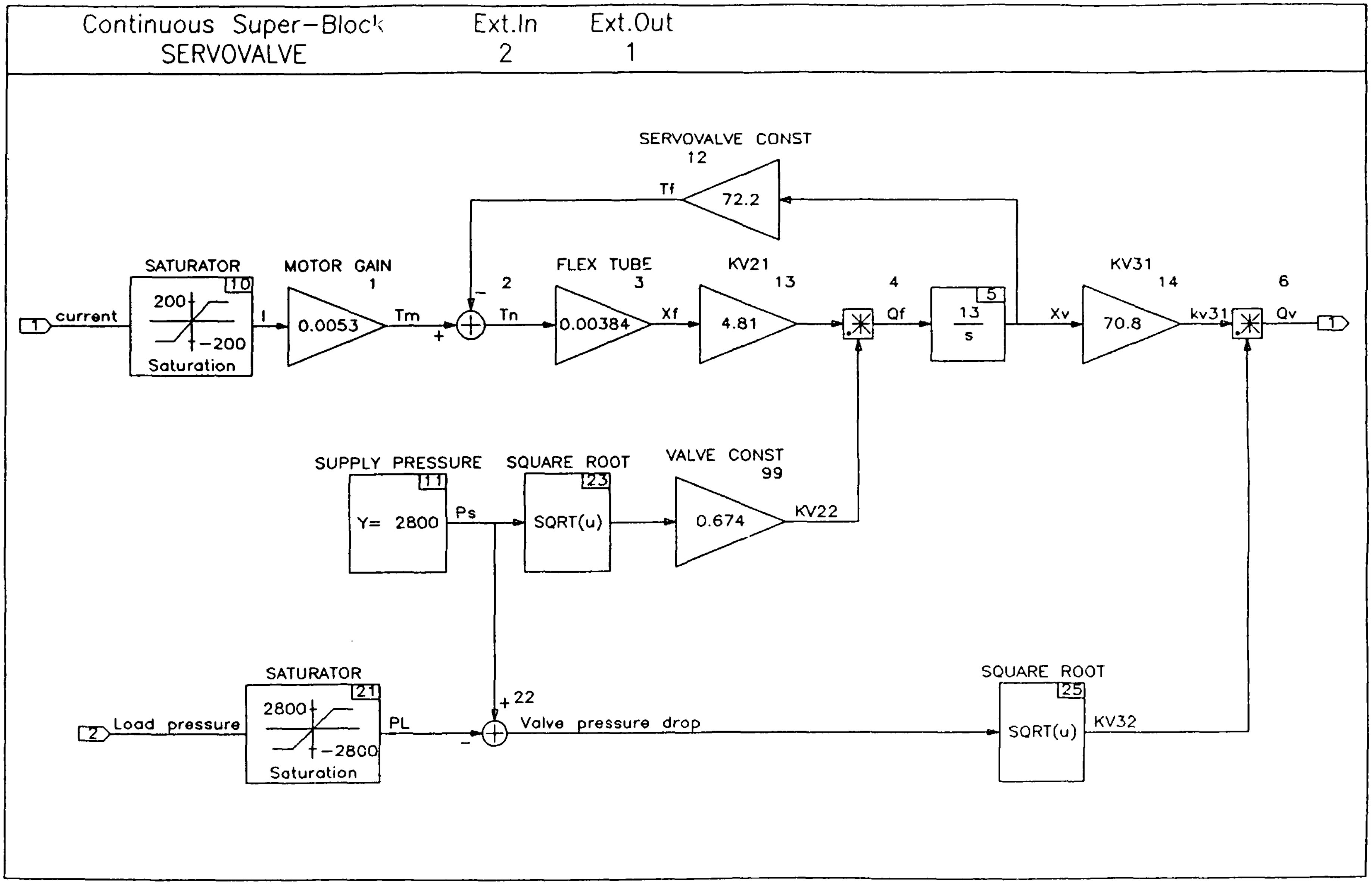


Figure 17
Servovalve computer model: super-block expansion.

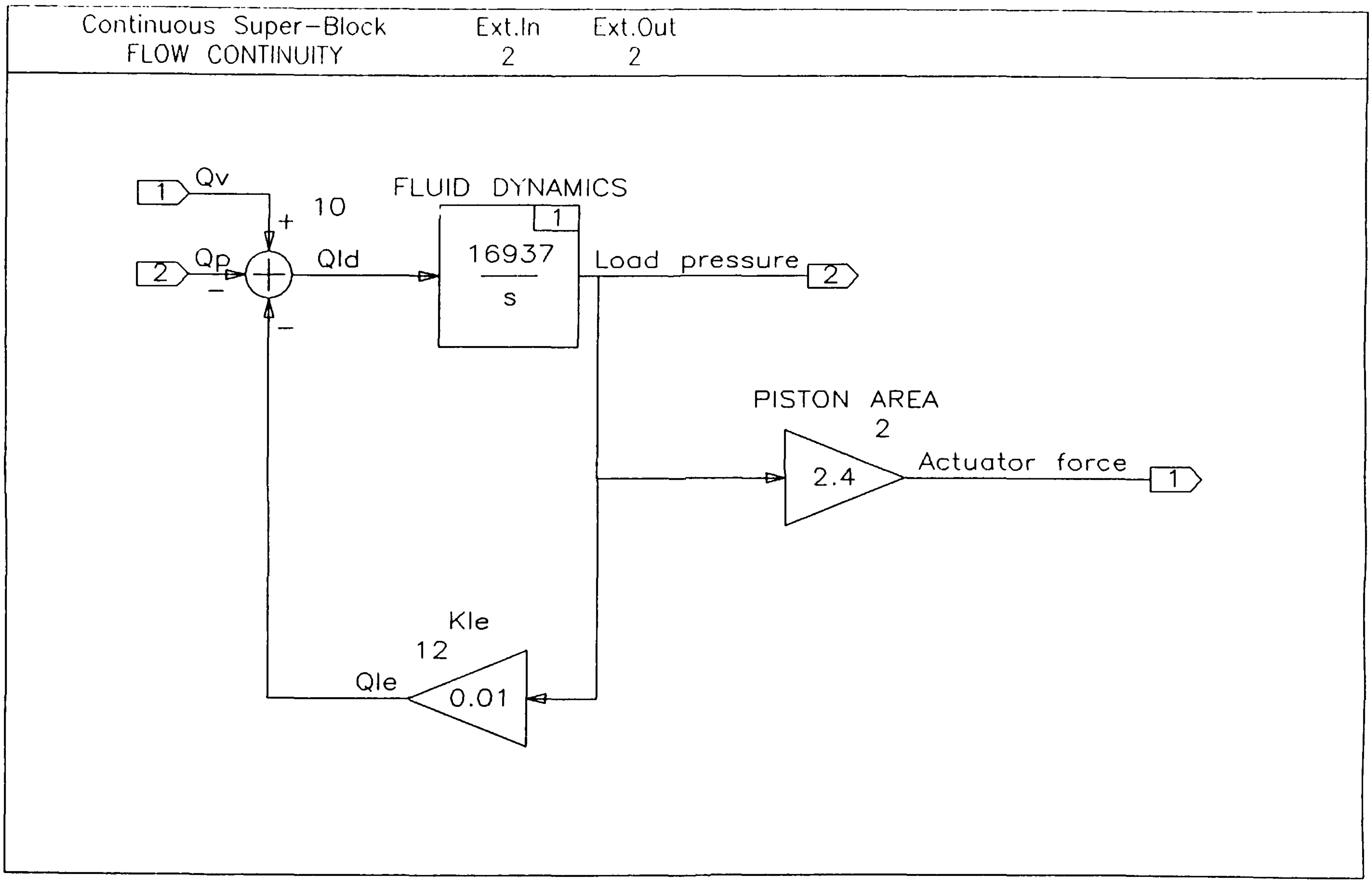


Figure 18
Flow continuity computer model: super-block expansion.

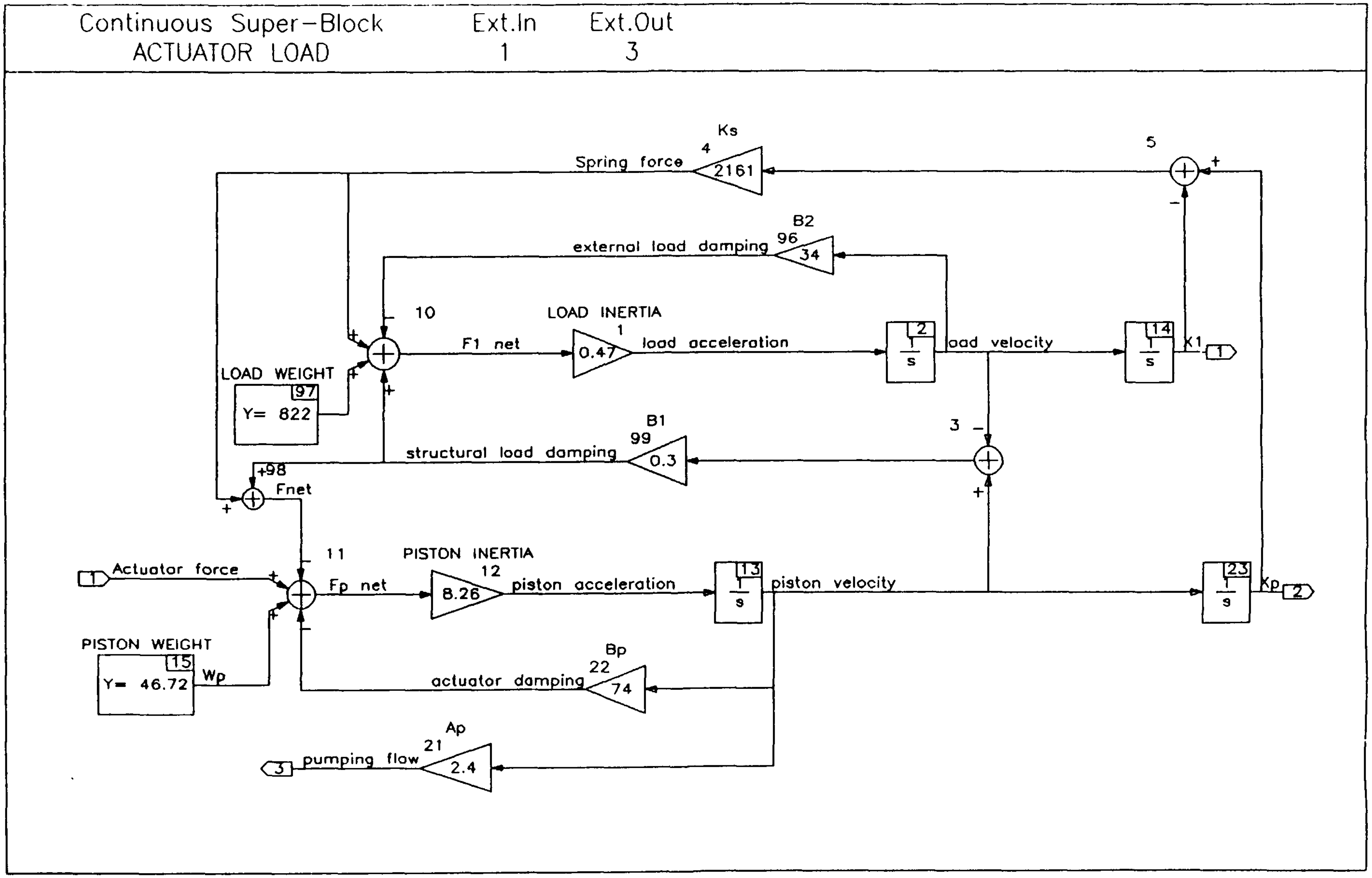


Figure 19
Baseline actuator/load computer model: super-block expansion.

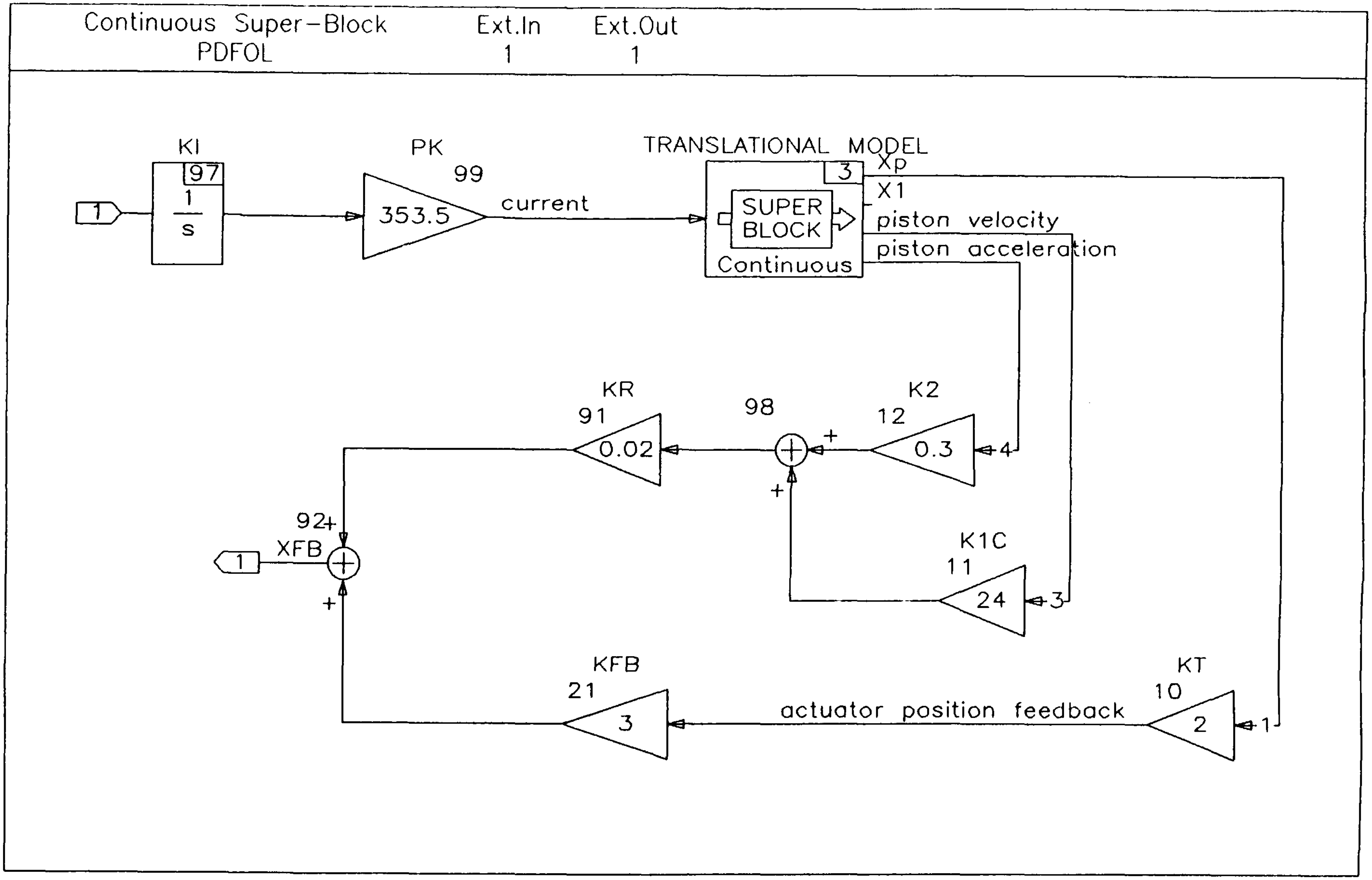


Figure 20
PDF open-loop system computer model.

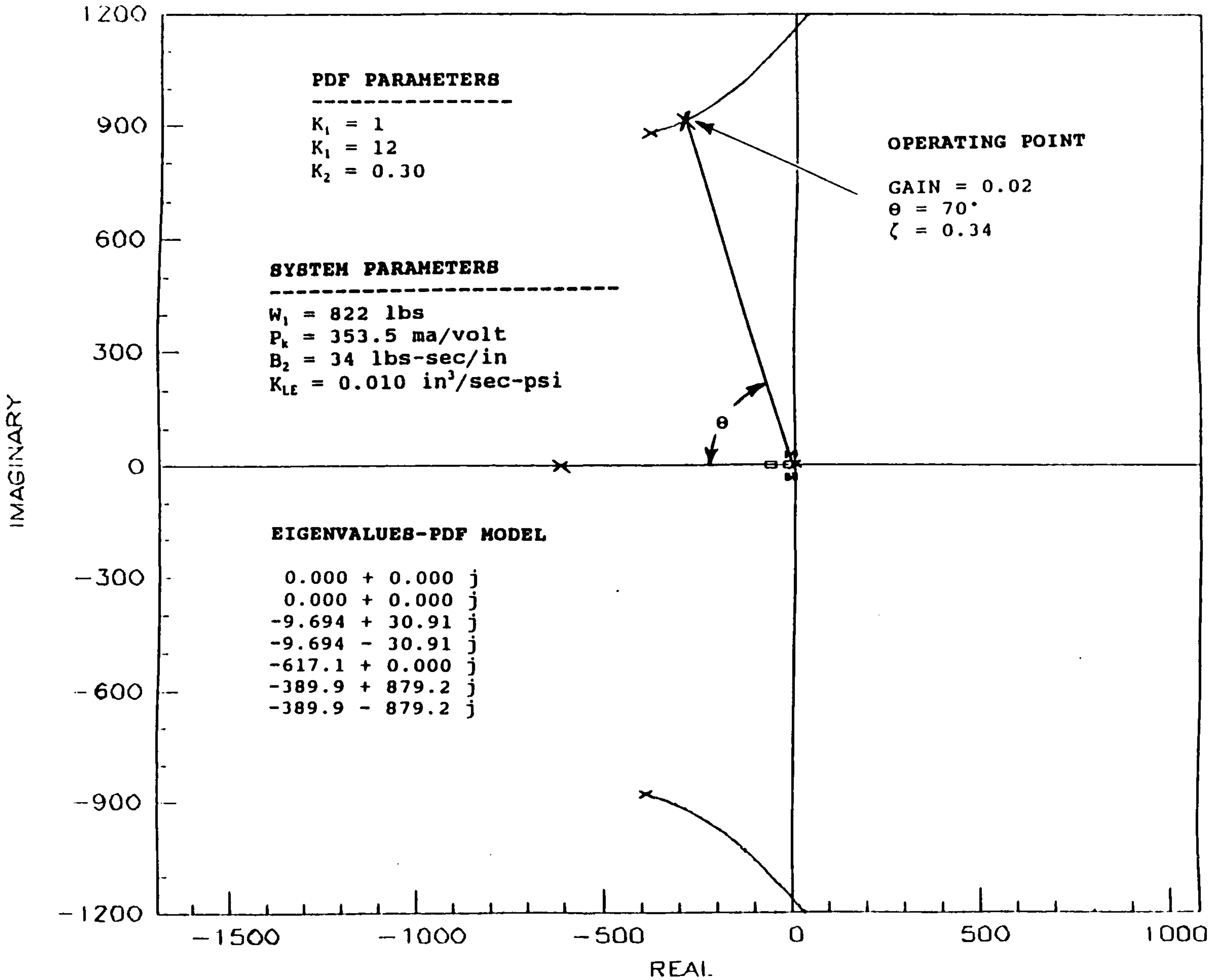


Figure 21
PDF system computer model: root locus.

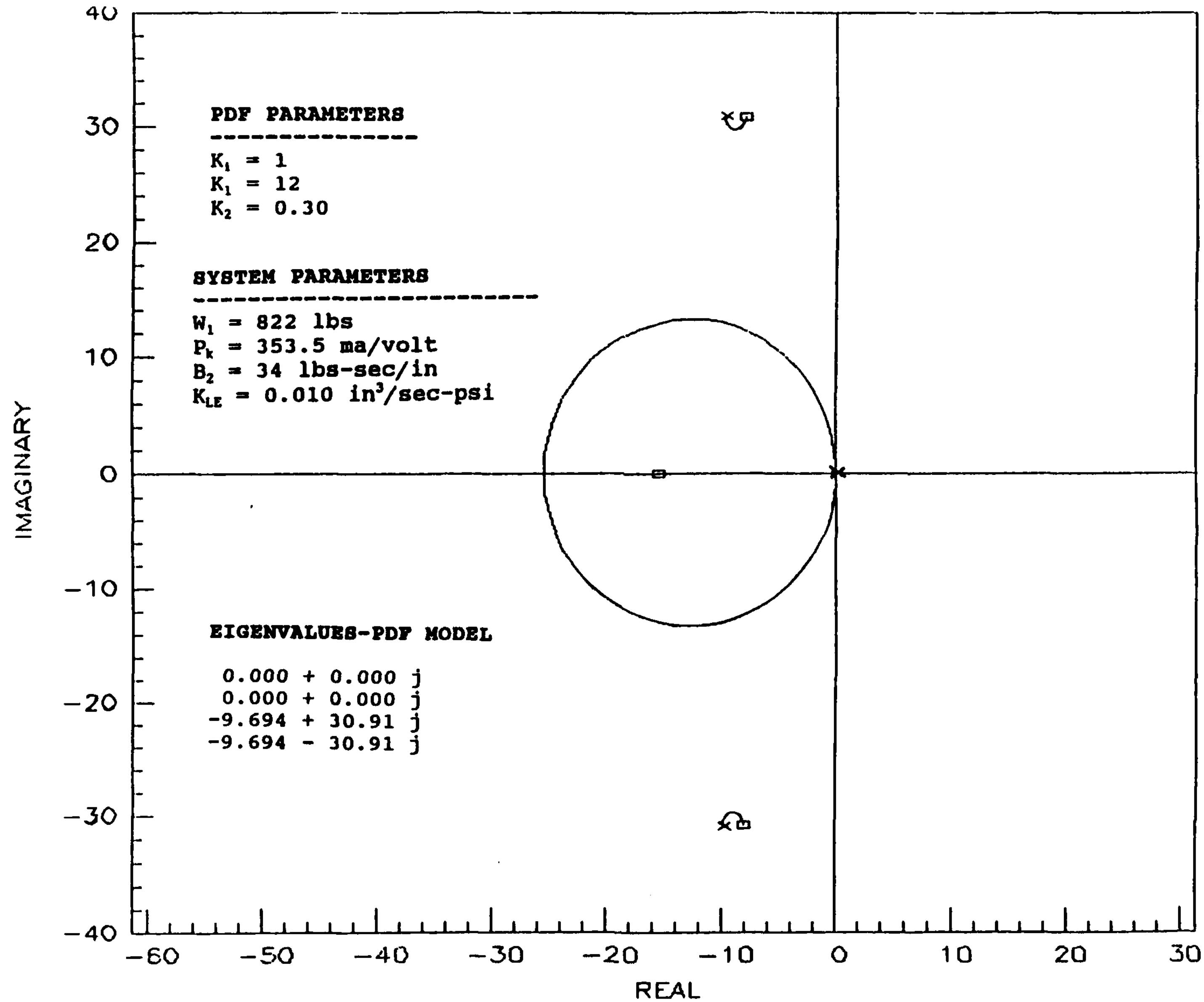


Figure 22
PDF system computer model: root locus, dominant roots.

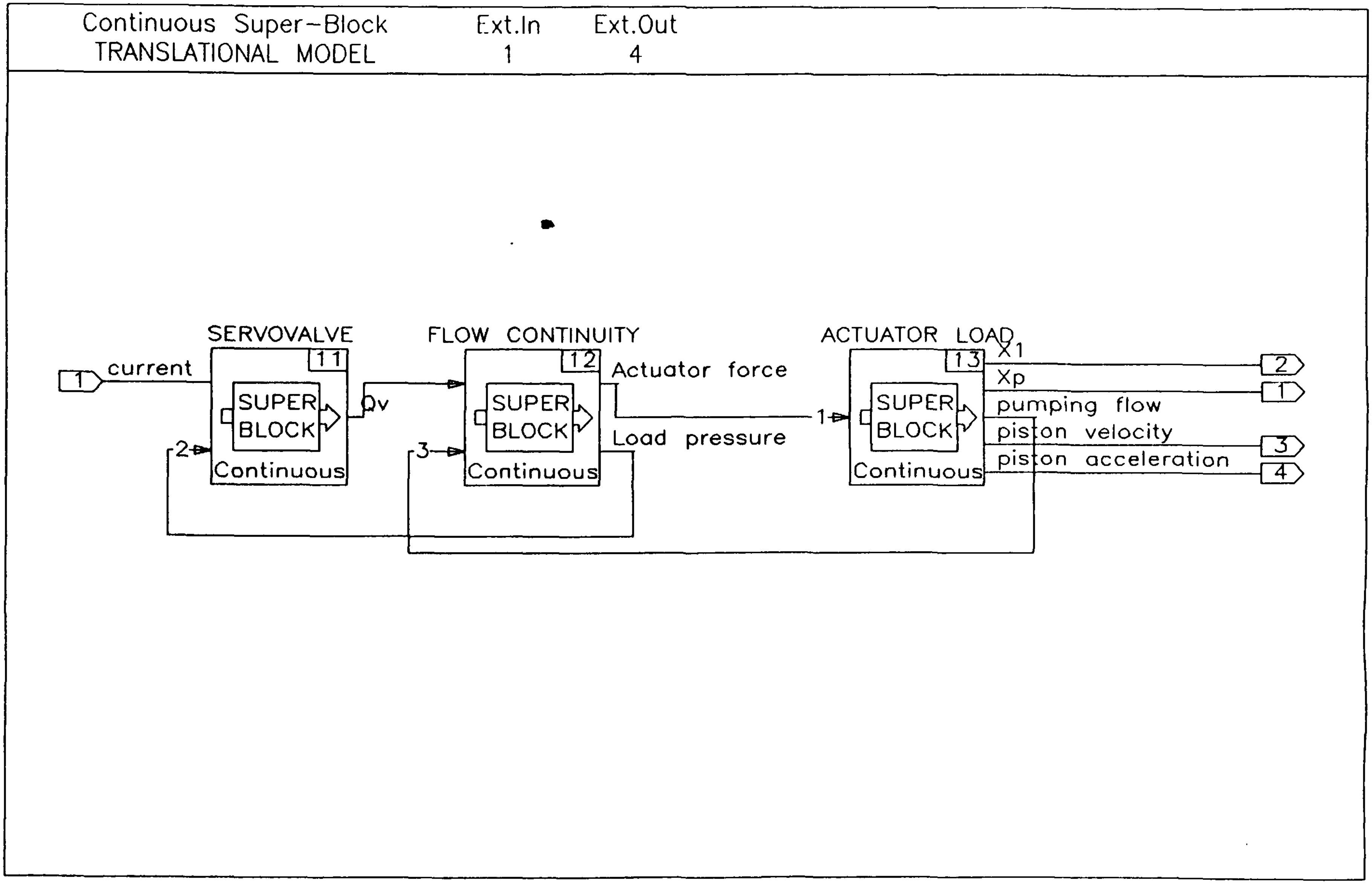


Figure 23
PDF translational model: super-block expansion.

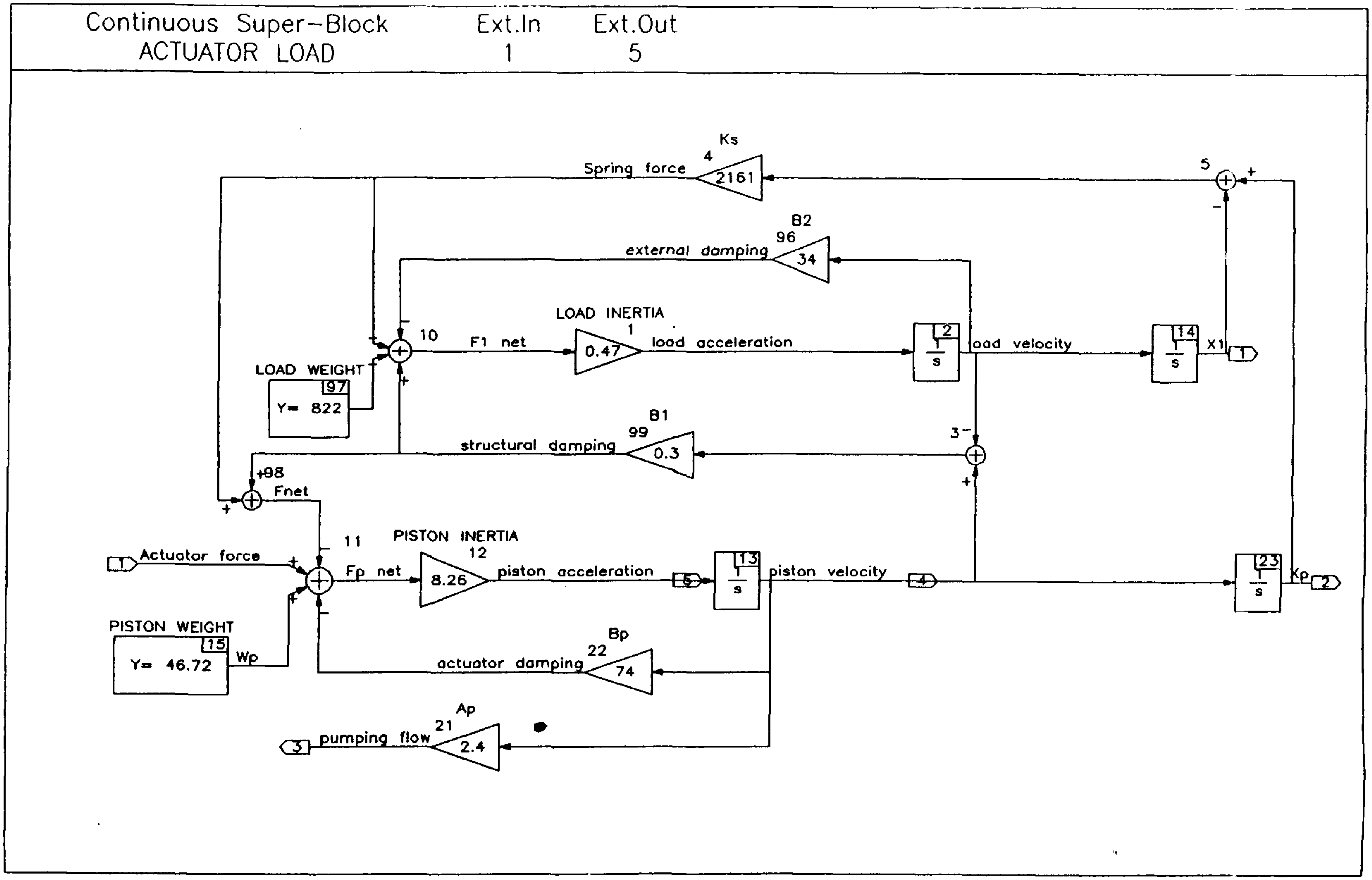


Figure 24
PDF actuator/load computer model: super-block expansion.

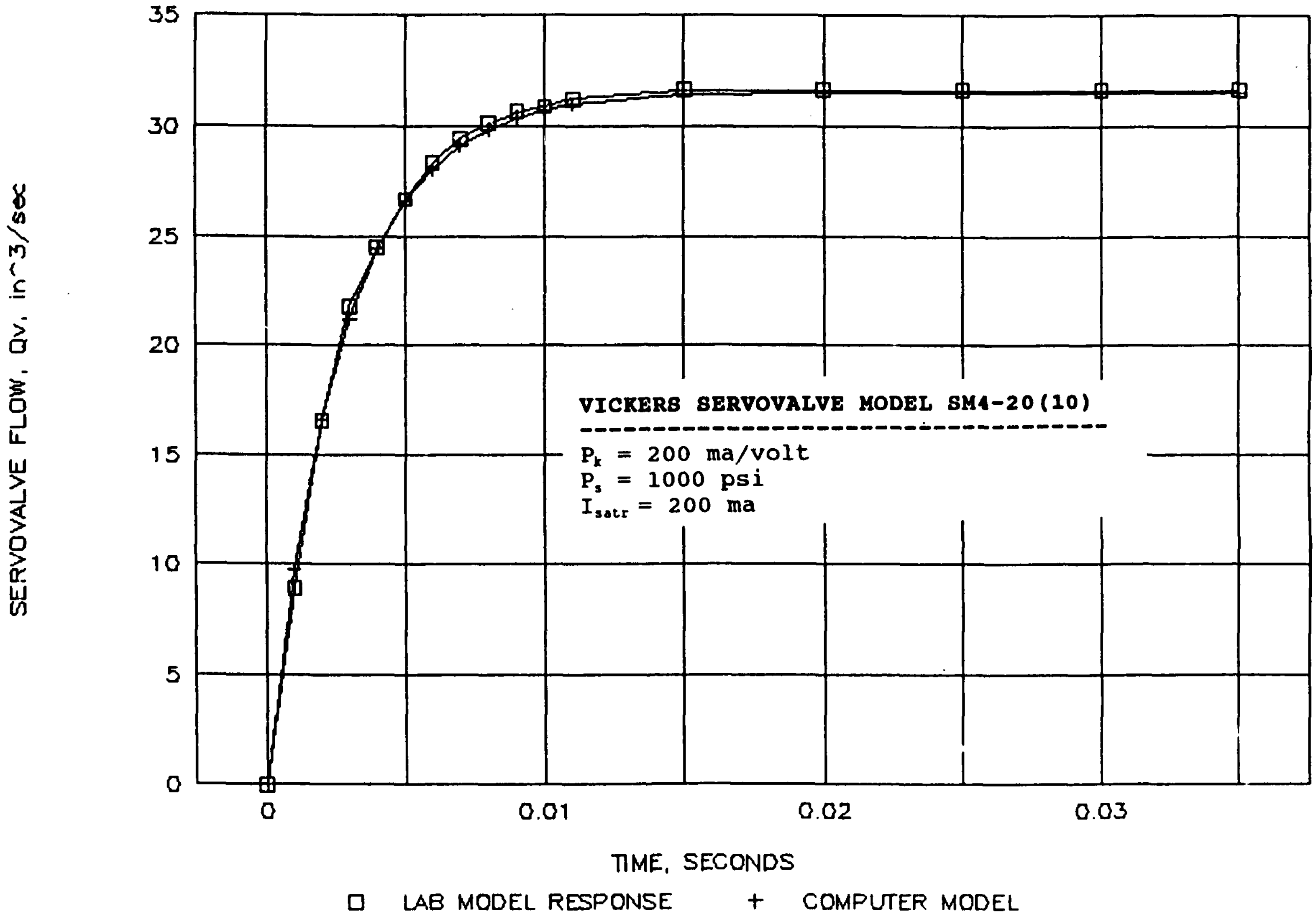


Figure 25
Servovalve model validation - step response.

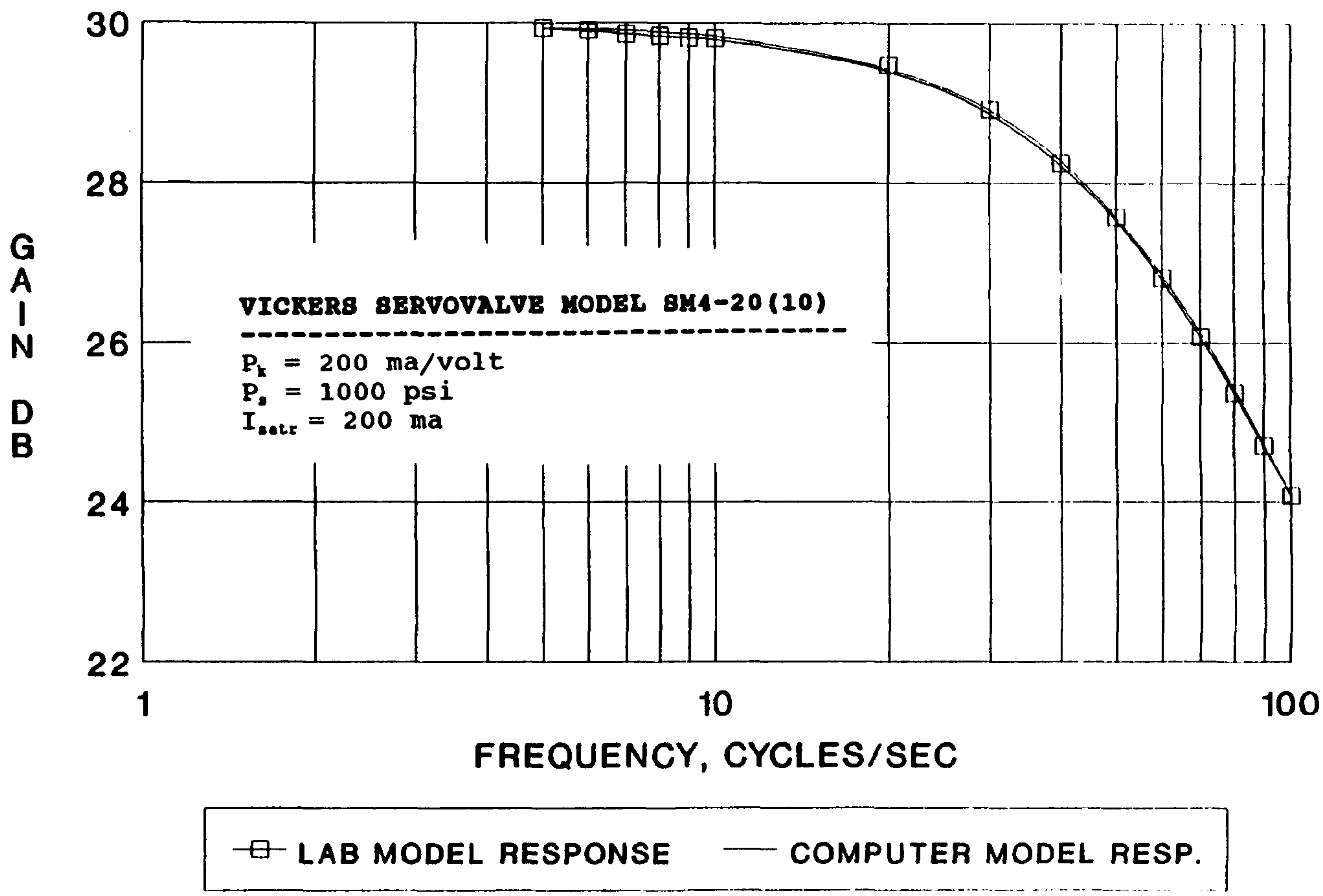


Figure 26
Servovalve model validation - frequency response.

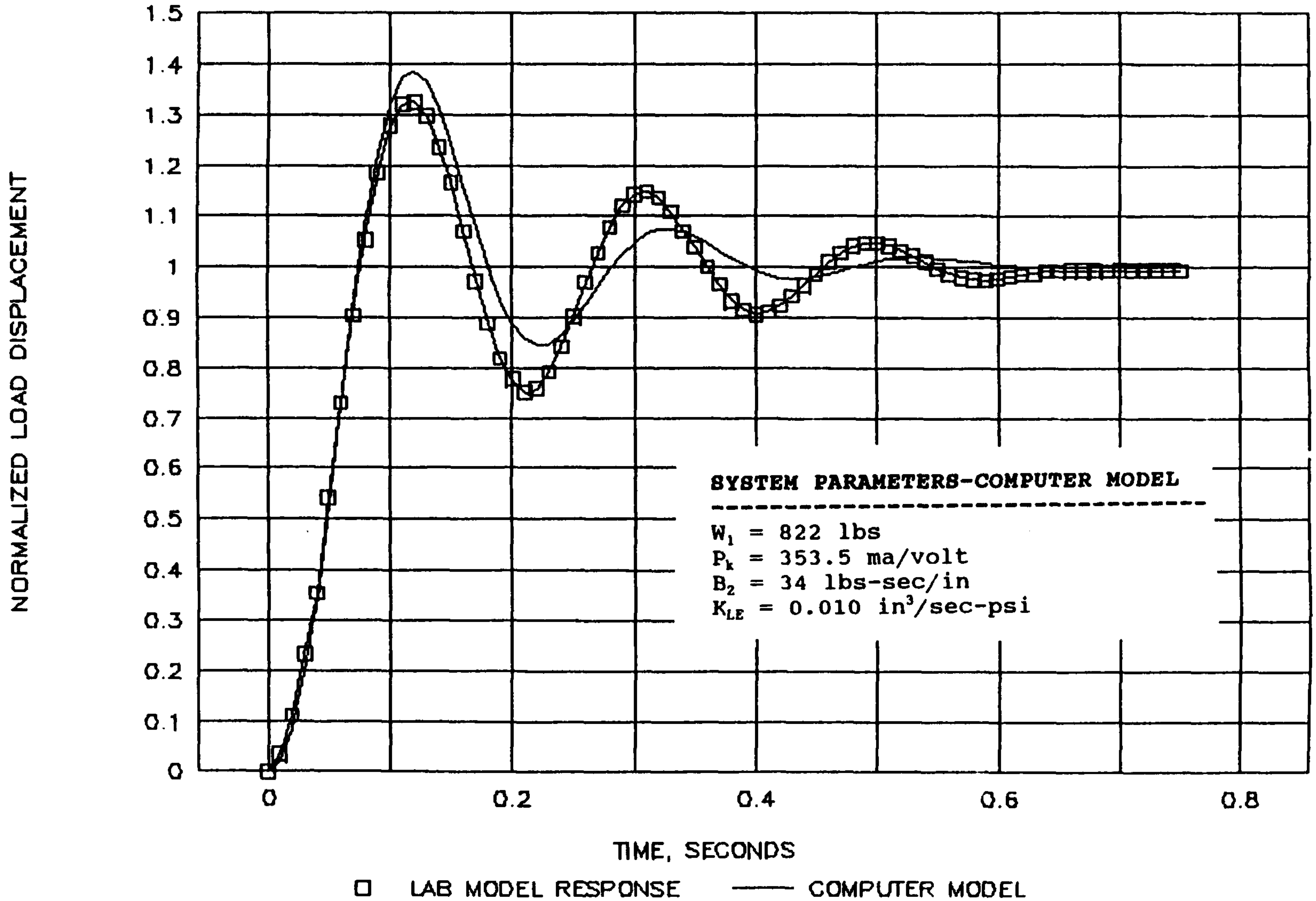


Figure 27
Baseline model validation - step response.

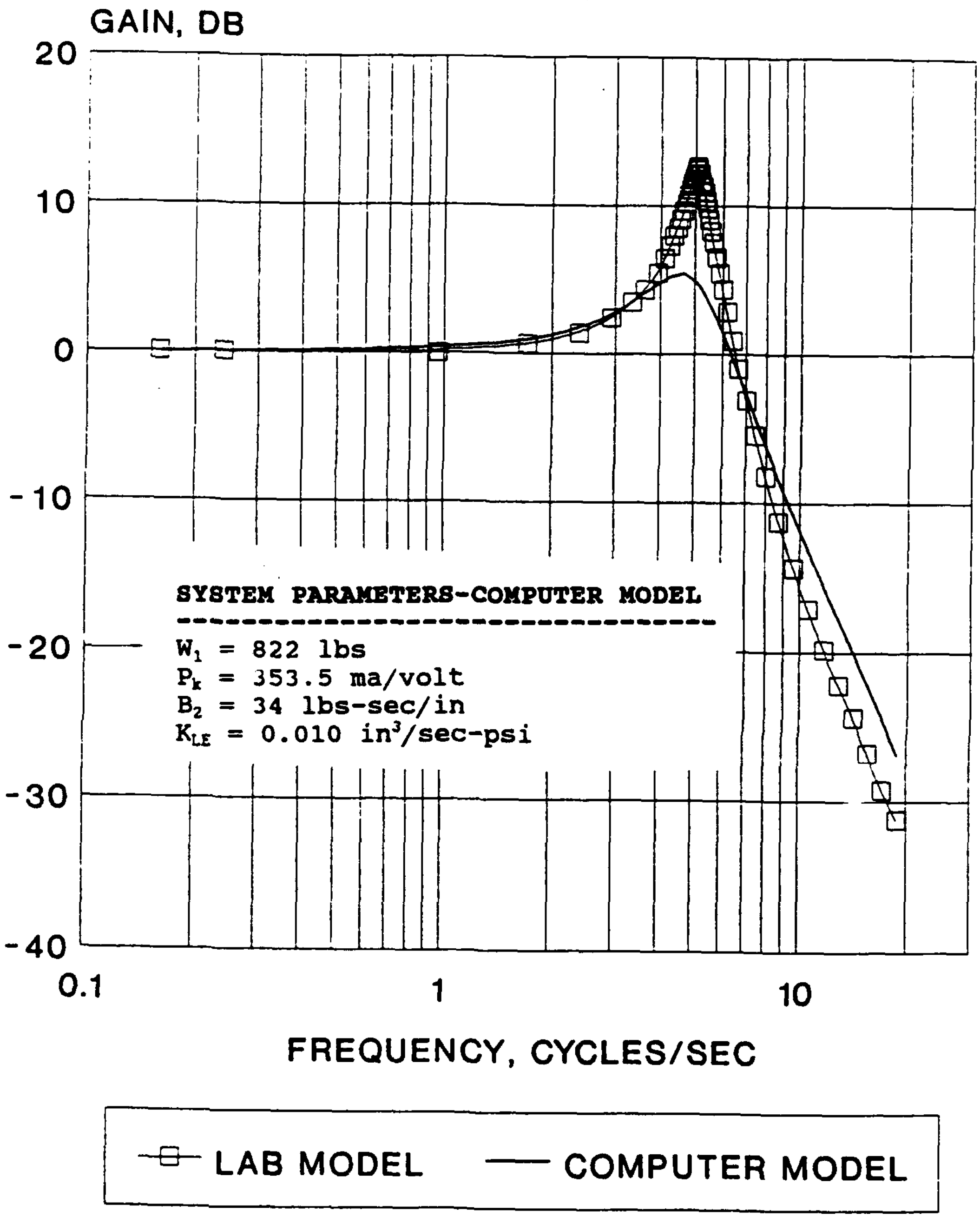


Figure 28
Baseline model validation - frequency response.

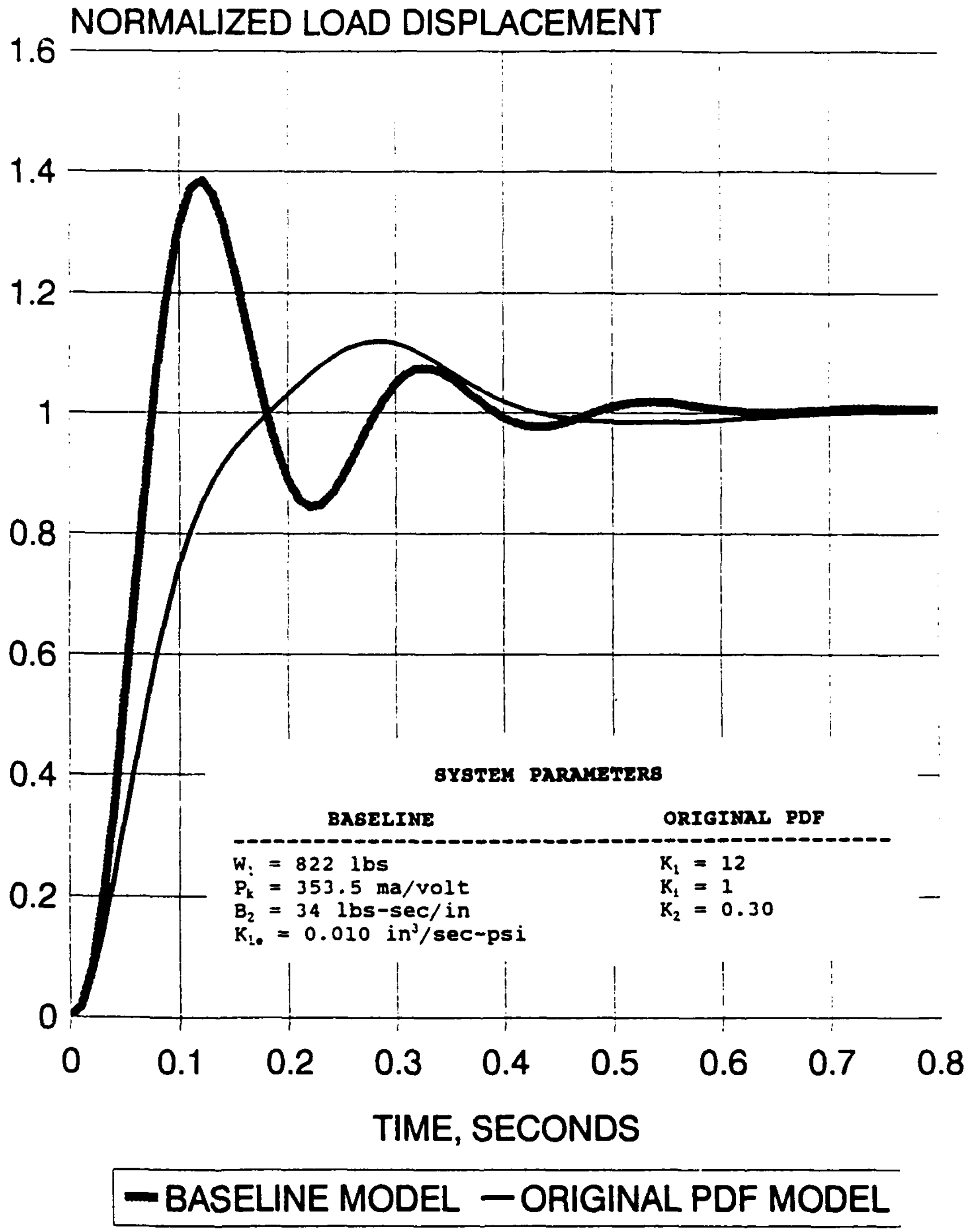


Figure 29
Step responses - original (nonoptimal) PDF and baseline computer models.

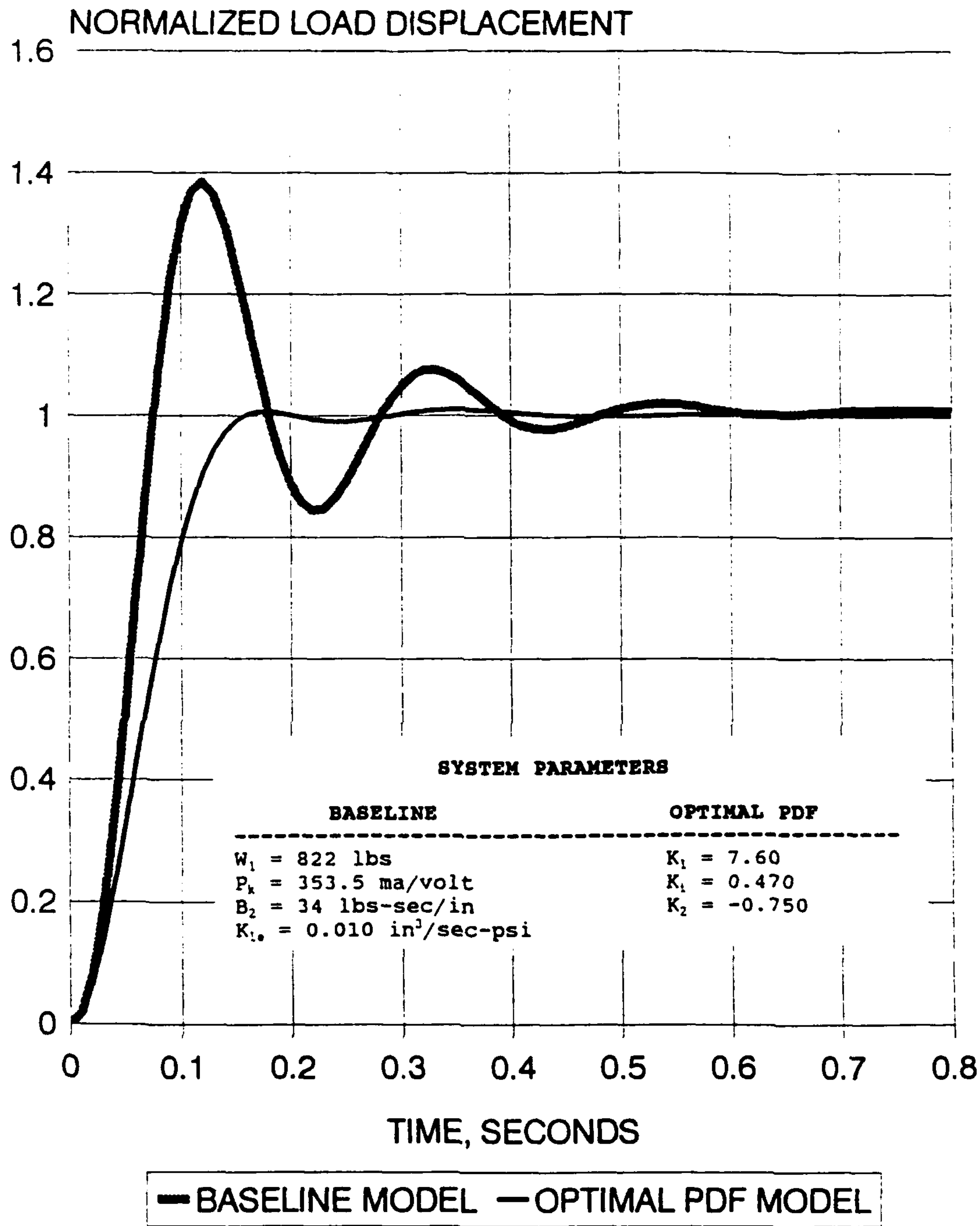


Figure 30
Step responses - optimal (manually tuned) PDF and baseline computer models.

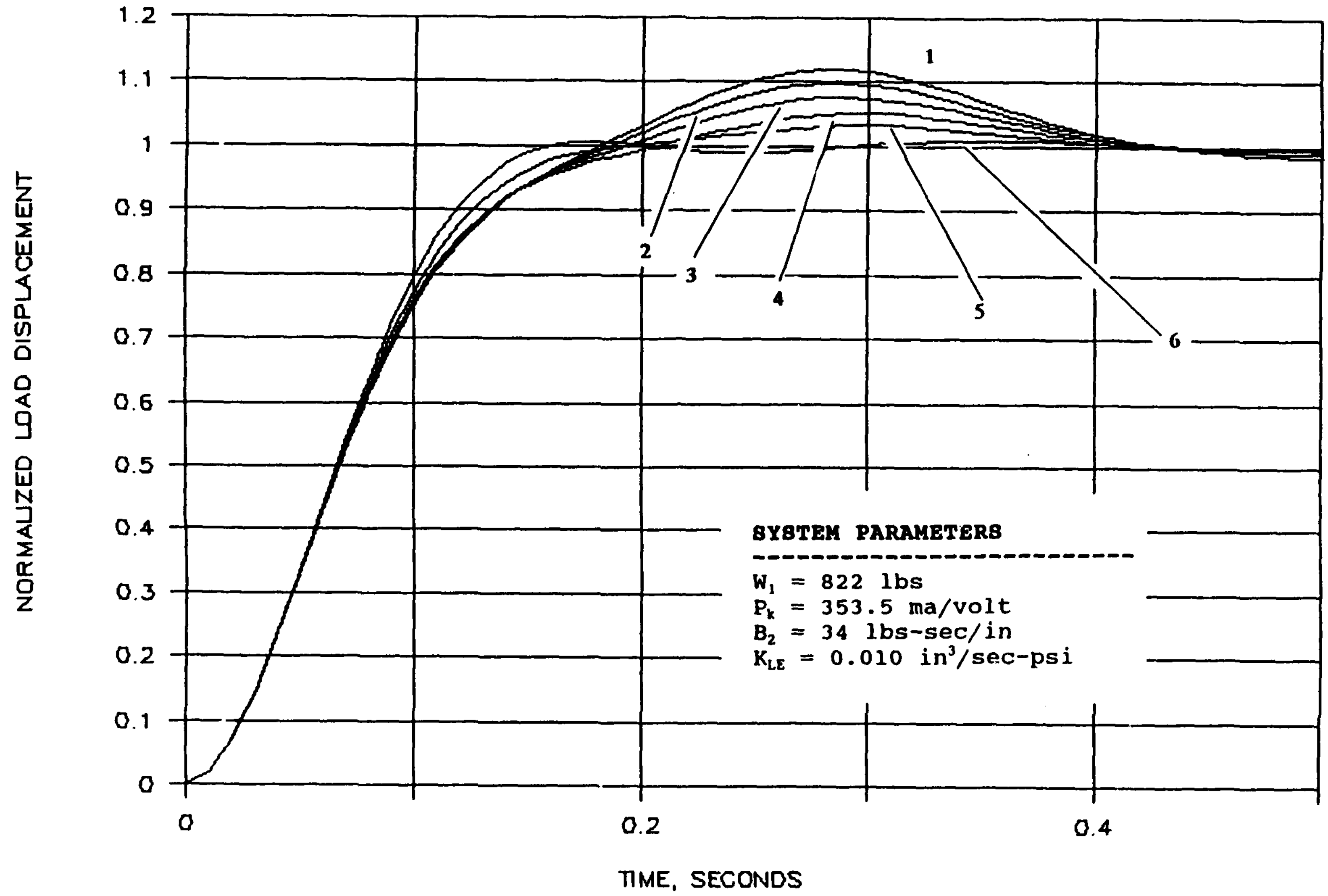


Figure 31
Convergence of computer optimize solution, case no. 1.

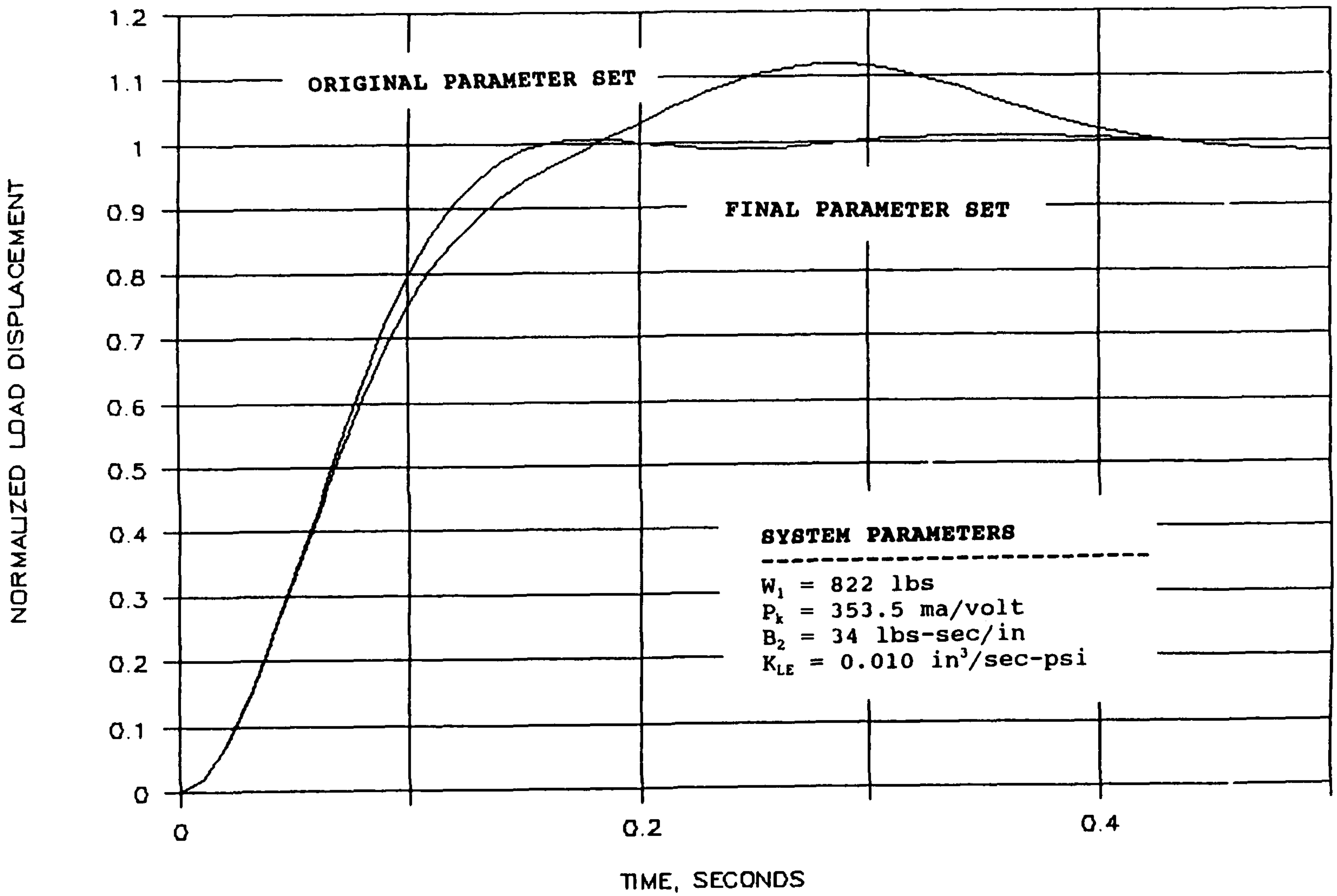


Figure 32
Step responses - original and final parameter sets, case no. 1.

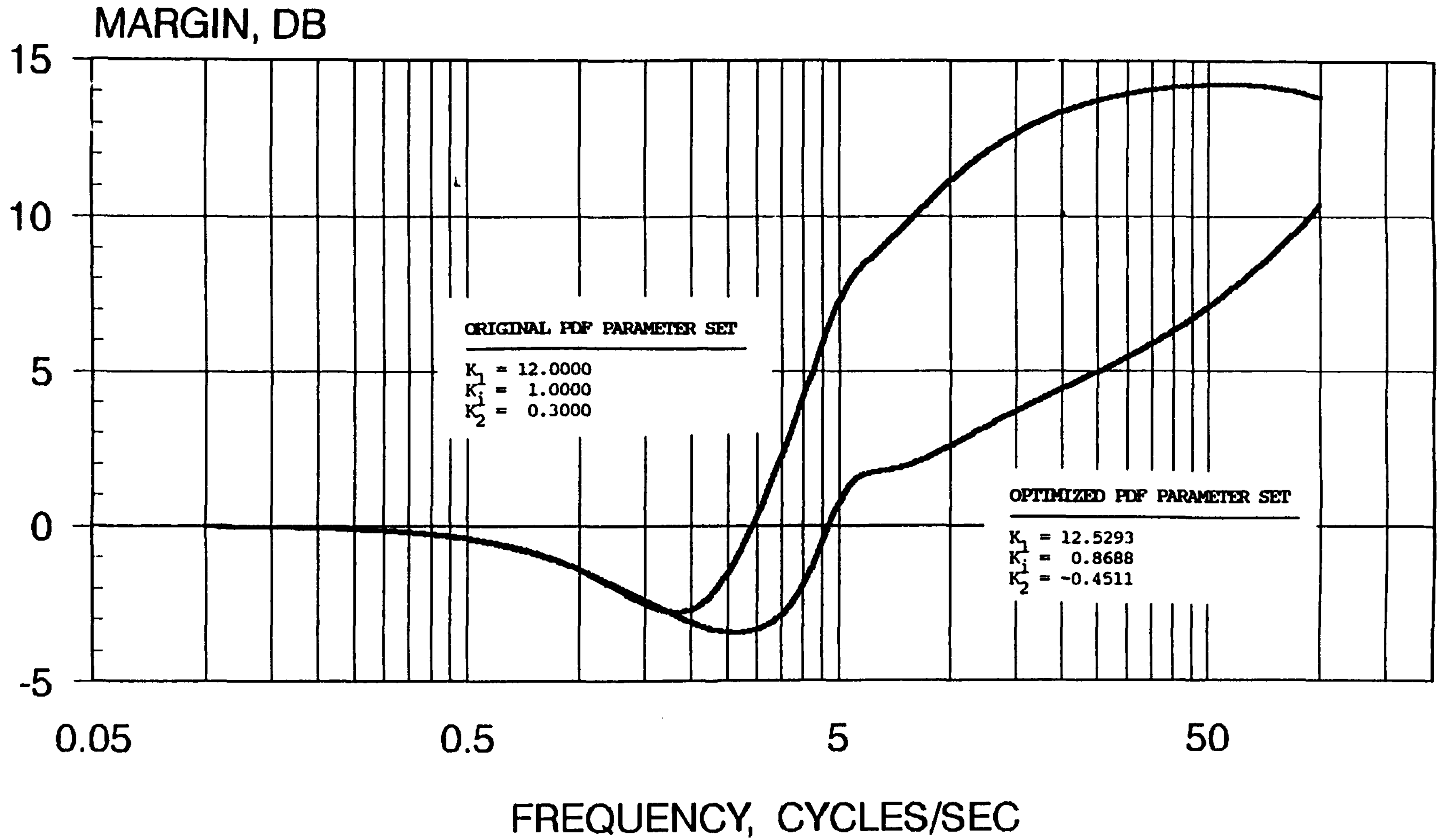


Figure 33
Multiplicative robustness margin, case no. 1.

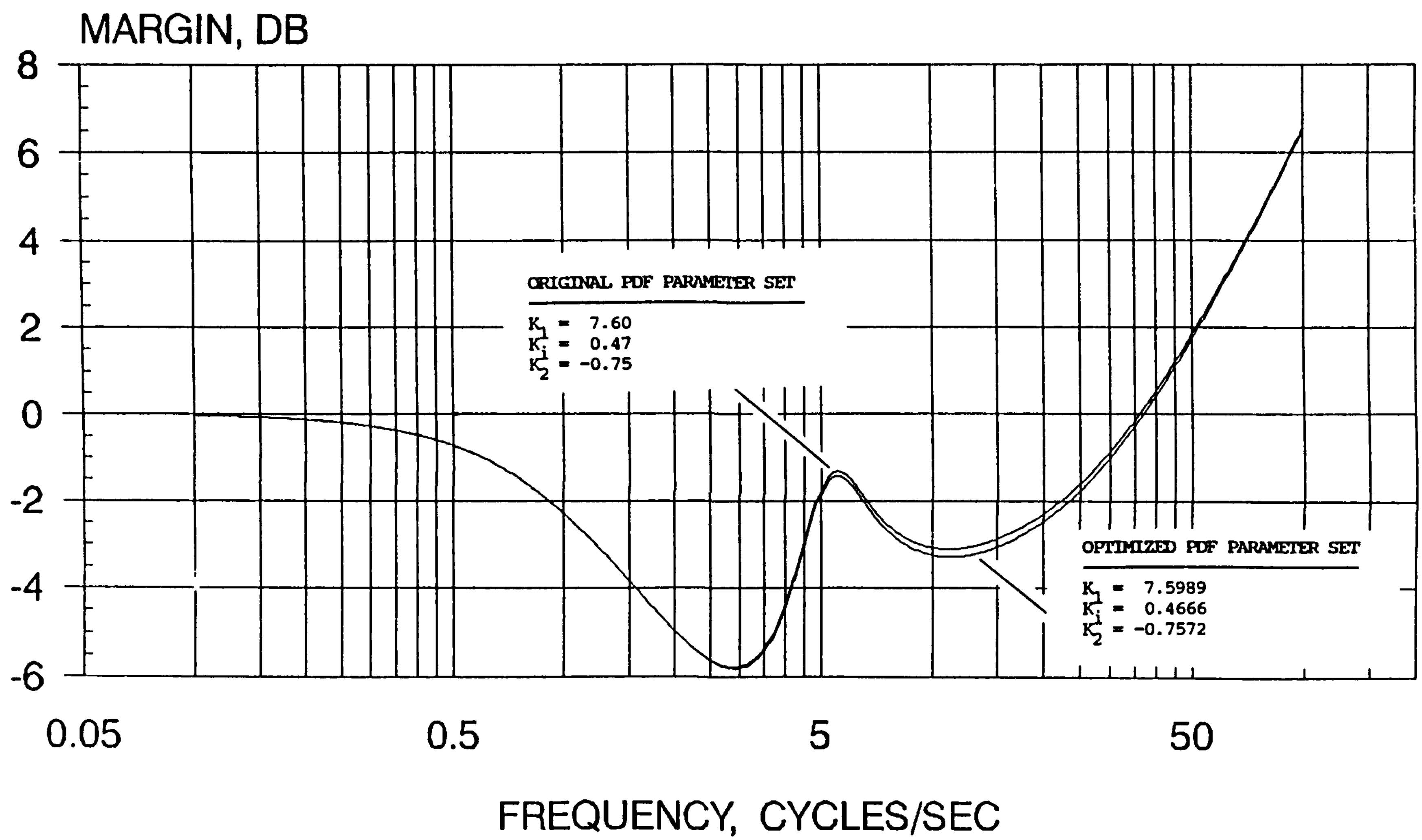


Figure 34
Multiplicative robustness margin, manually tuned system.

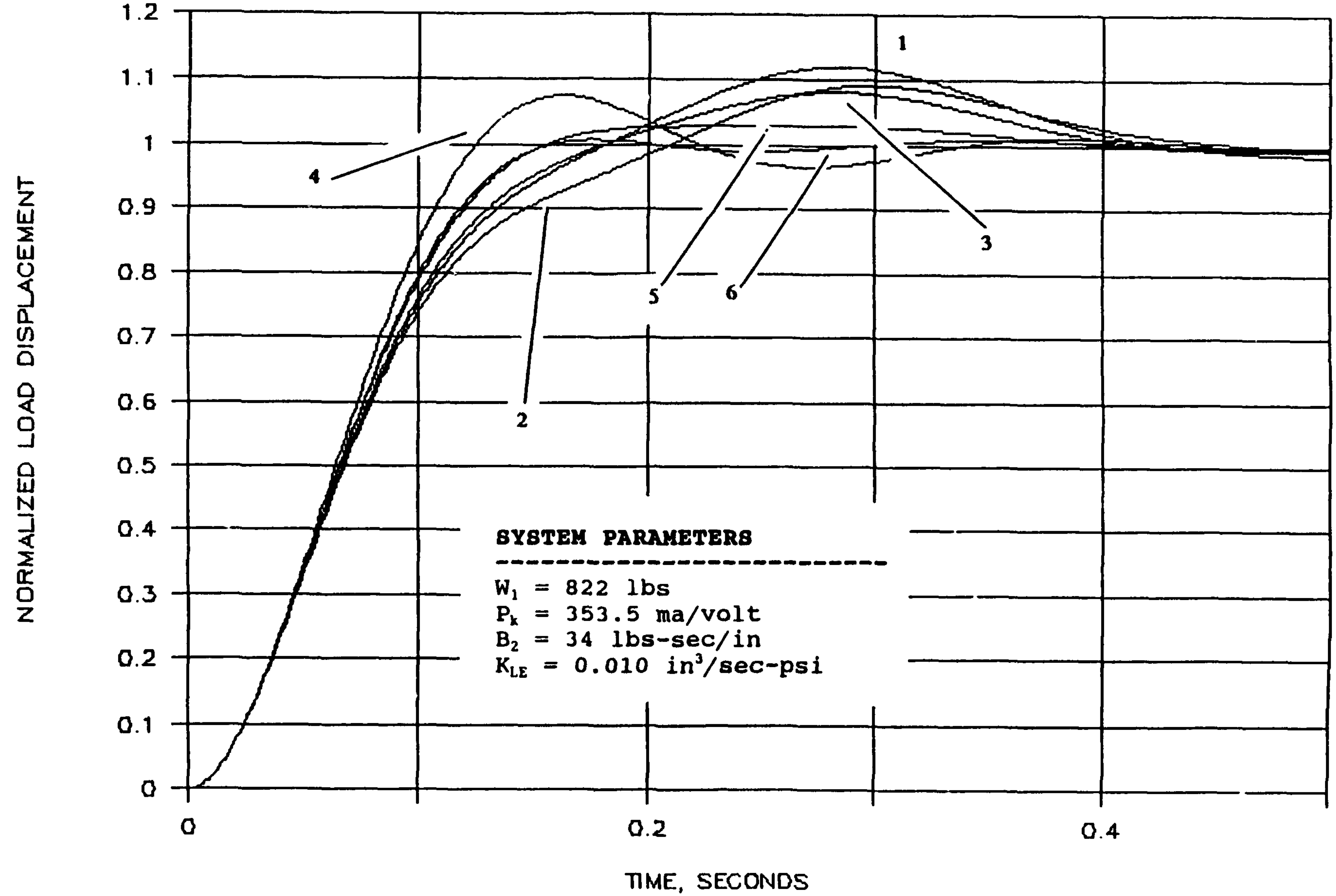


Figure 35
Convergence of computer optimize solution, case no. 2.

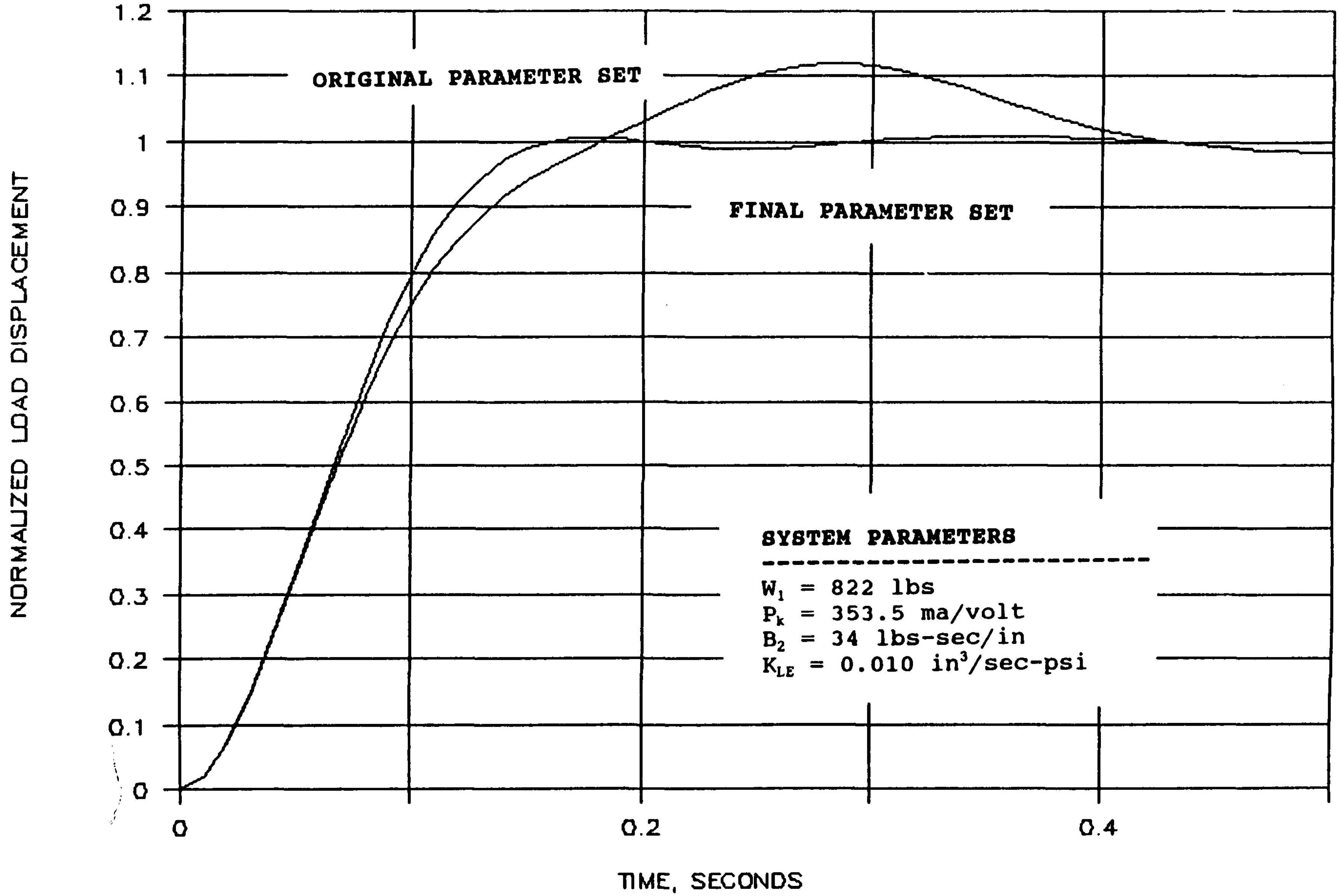


Figure 36
Step responses - original and final parameter sets, case no. 2.

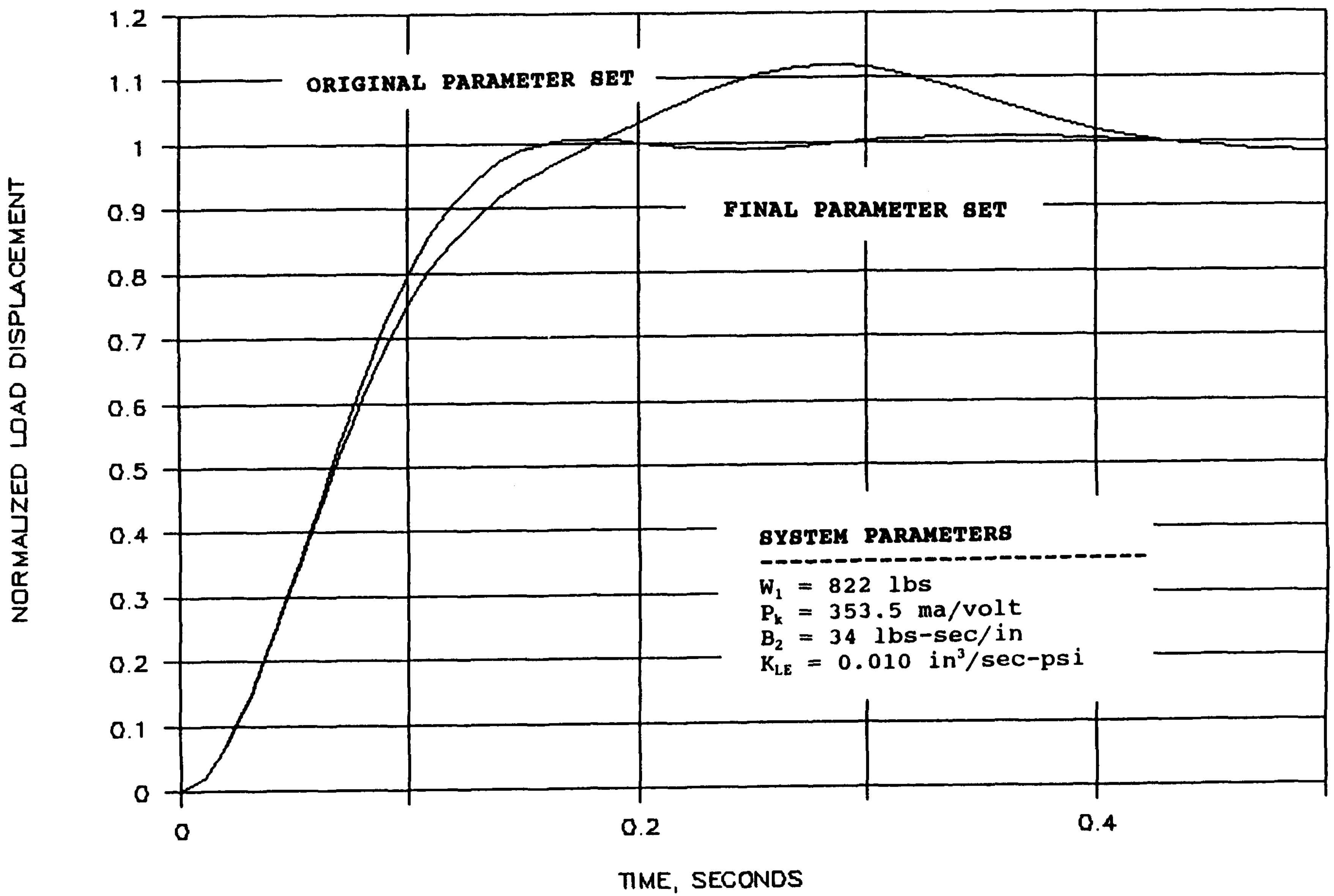


Figure 37
Step responses - original and final parameter sets, case no. 3.

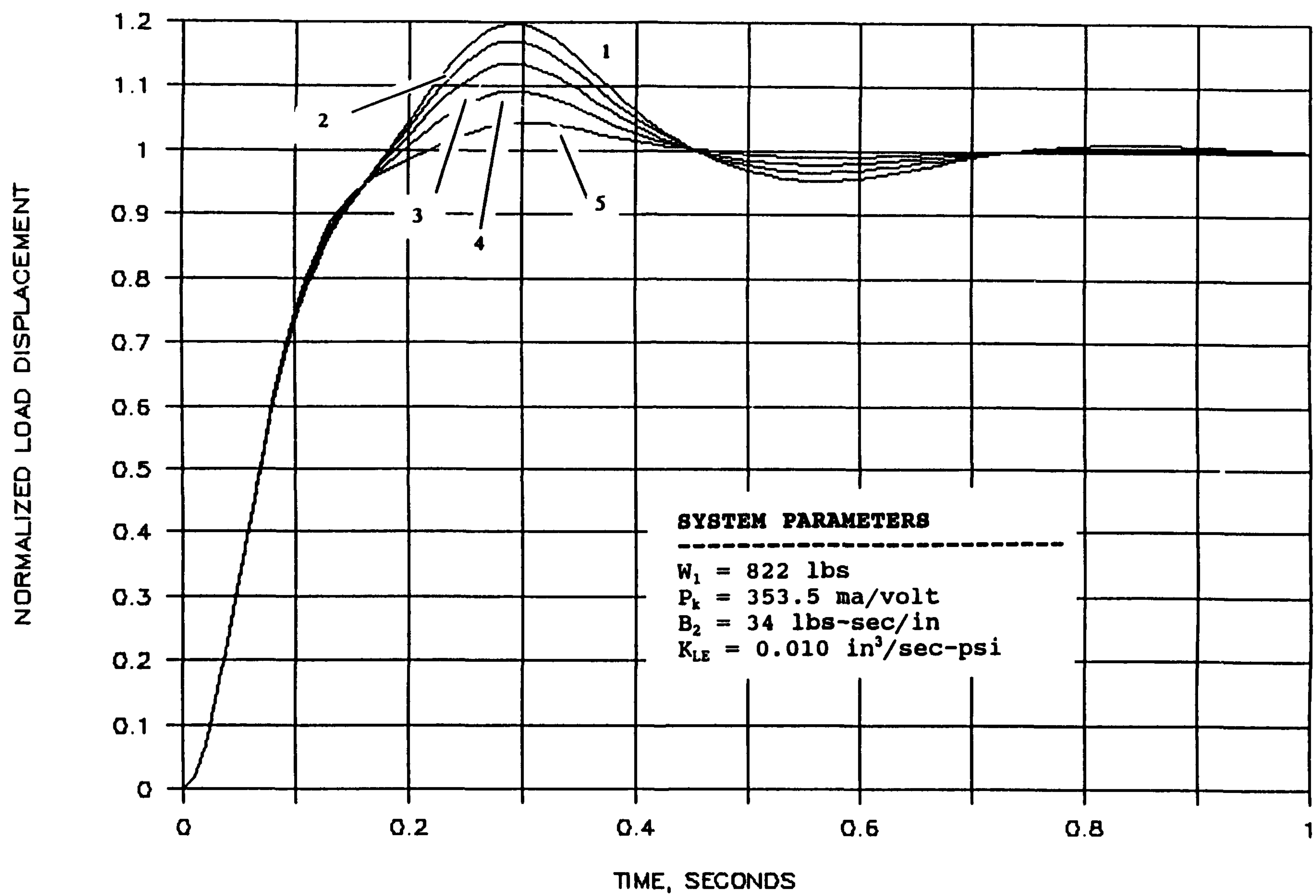


Figure 38
Convergence of computer optimize solution, case no. 4.

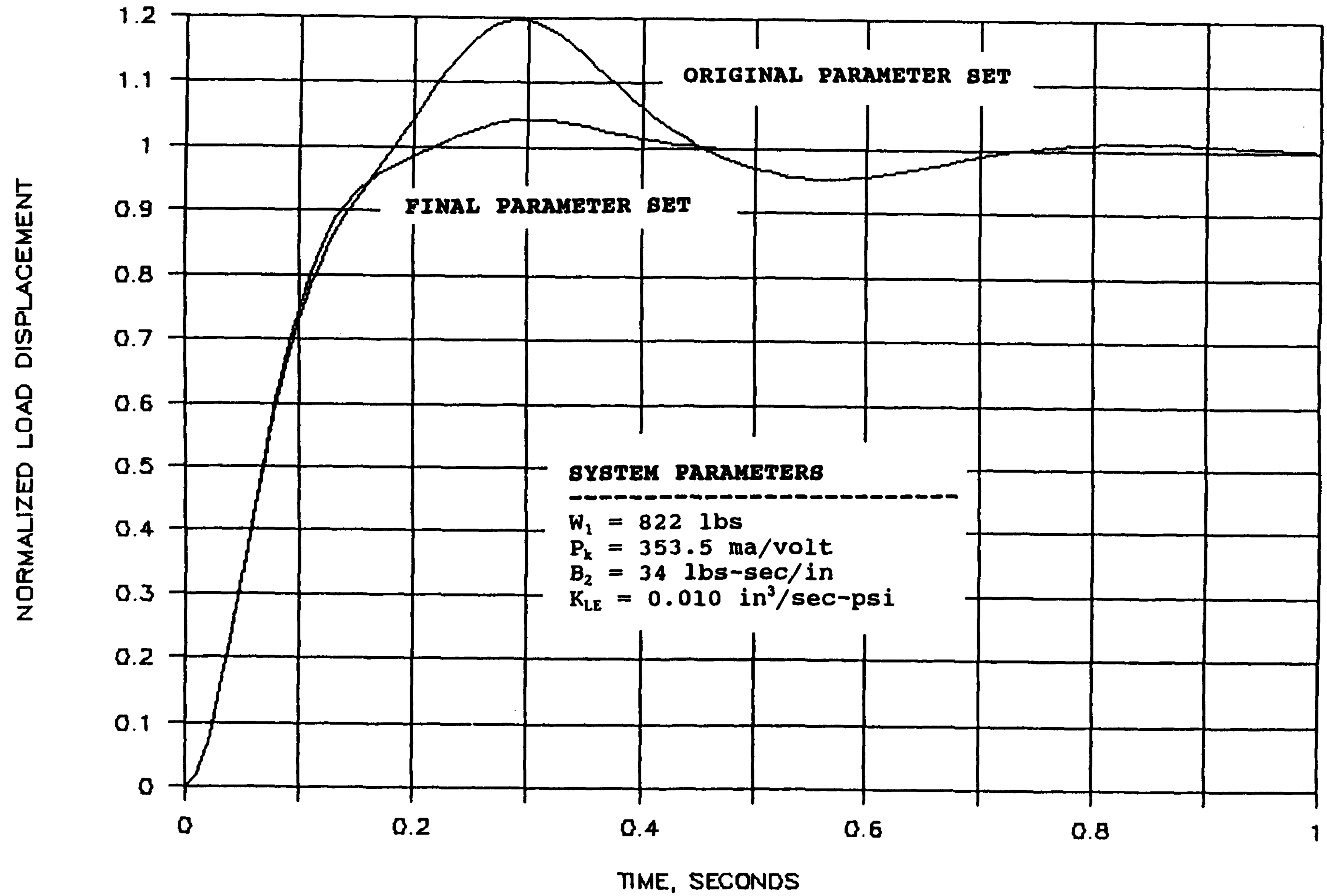


Figure 39
Step responses - original and final parameter sets, case no. 4.

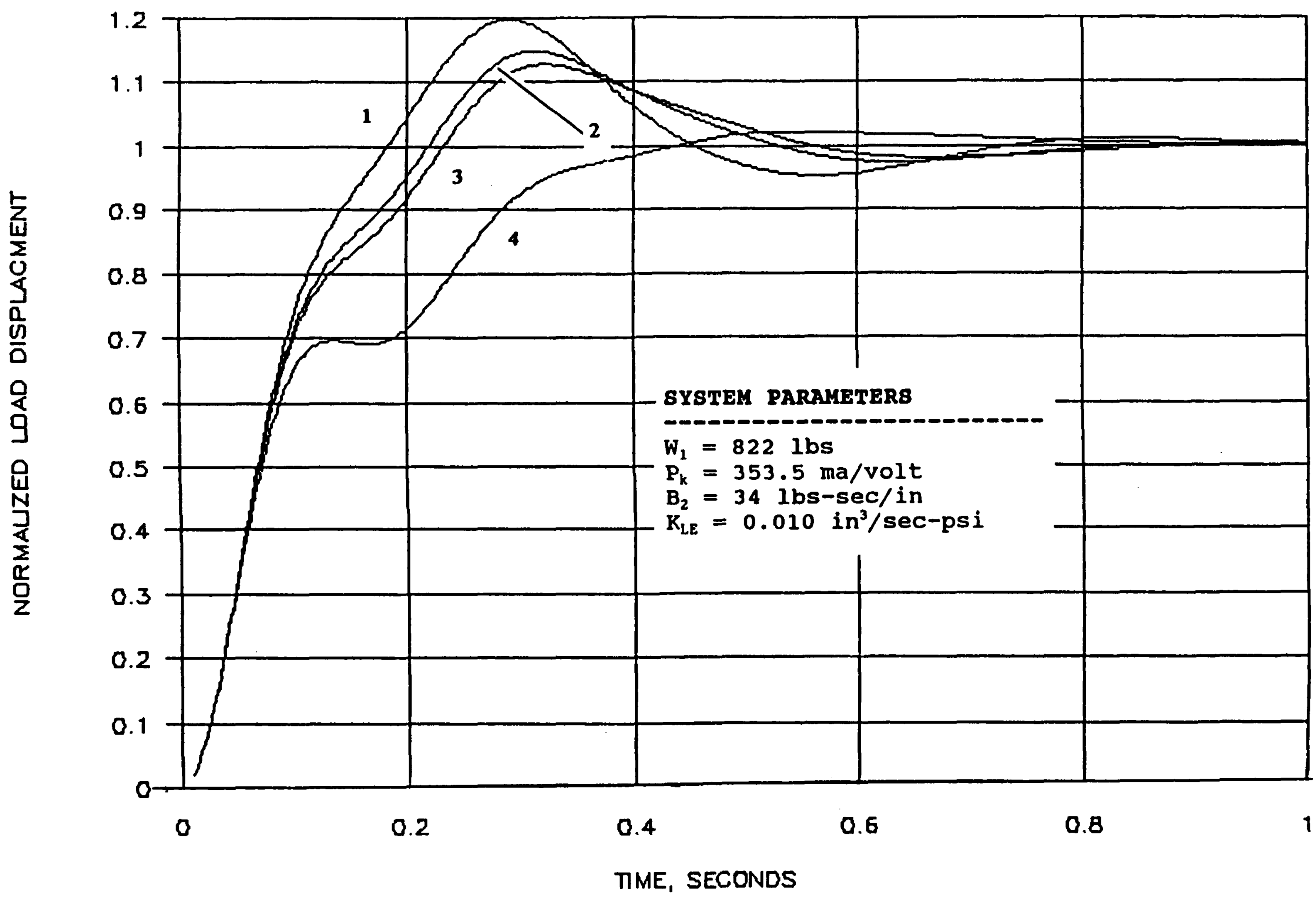


Figure 40
Convergence of computer optimize solution, case no. 5.

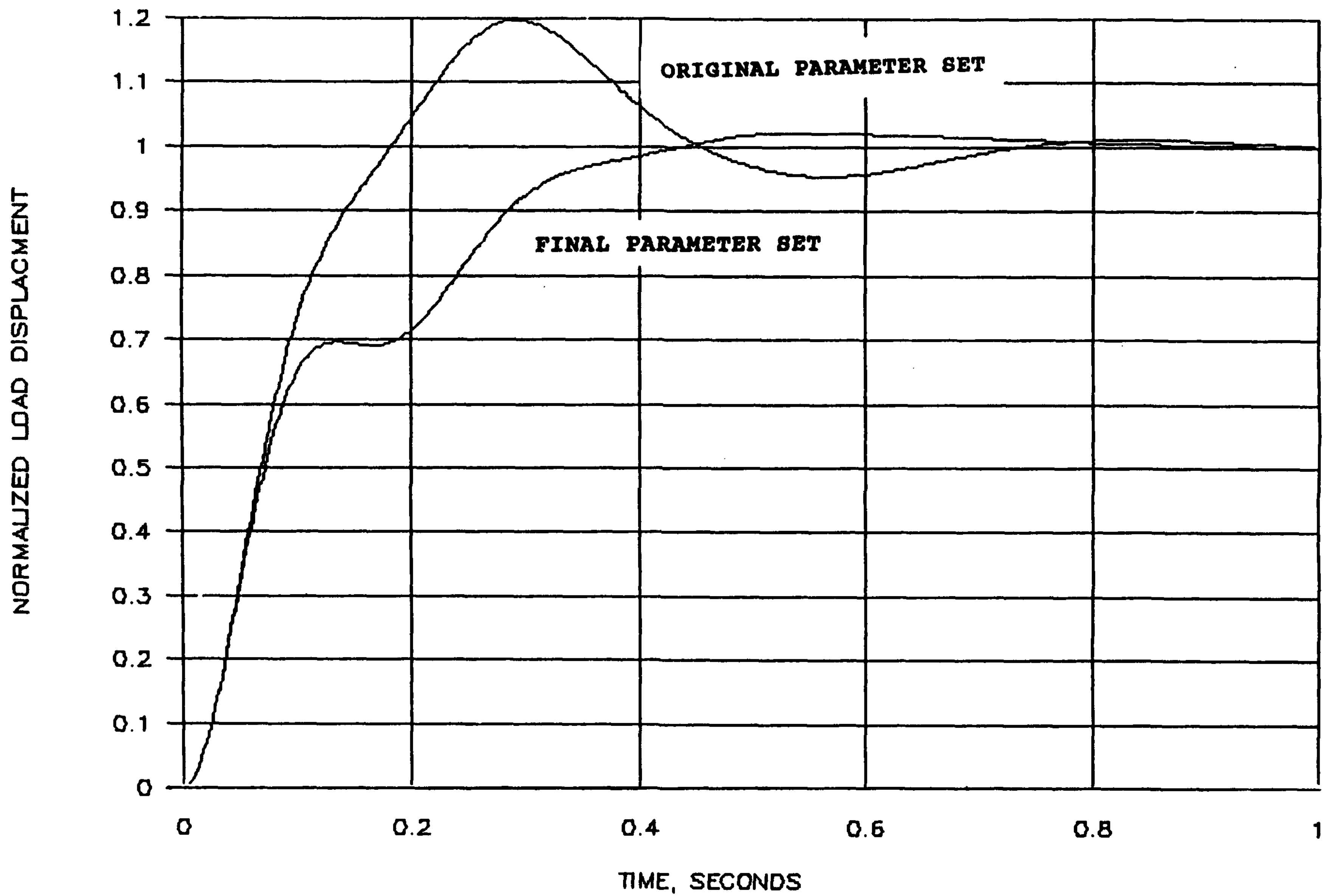


Figure 41
Step responses - original and final parameter sets, case no. 5.

Appendix A

LABORATORY MODEL COMPONENT DESCRIPTIONS

HYDRAULIC ACTUATOR

Instron model (catalog) No. 3375-1050-2-5 Electrohydraulic Actuator, with integrated Linear Variable Displacement Transducer (LVDT).

SERVOVALVE

Vickers model SM4-20 (10 US GPM) Electrohydraulic Servovalve.

SERVOCONTROLLER

Vickers model EMD-30 Servo Amplifier with Proportional, Integral and Derivative (PID) (feedback), modified for proportional only or PDF.

SERVOCONTROLLER POWER SUPPLY

Vickers Model EMP-A-20 Power Supply.

LOAD CELL

Instron Series 2518 Load Cell, calibrated at 5,000 Kilo-newtons/Volt.

SIGNAL GENERATOR

Wavetek Model 132 VCG/Noise Generator.

SIGNAL ANALYZER

Hewlett-Packard Model HP35665A Signal Analyzer.

Appendix B

SERVOVALVE FLOW CONSTANT CALCULATIONS

INTRODUCTION

The servovalve used in the laboratory model of the backhoe boom position control system was used to control the hydraulic actuator. The technical requirements for the servovalve, which were specified by NCEL in the statement of work, were a flow rate of 10 gpm at a supply pressure of 1,000 psi, and a pressure rating of 3,000 psi. A Vickers model SM4-20(10) servovalve was selected by TS&S to meet these requirements.

Two servovalve flow constants, K_{v21} and K_{v31}, included in the servovalve block diagram, are a combination of several other system constants. The following calculations show how these two constants were developed.

PILOT STAGE FLAPPER CONSTANT

The input to the servovalve is electrical current which causes the torque motor to rotate which in turn moves the flapper to the right side against the nozzle. The flow associated with this is given by:

$$Q_f = K_{v21}K_{v22}X_f$$

where

X_f = flapper displacement, inches

The term K_{v21} is called the pilot stage flapper sensitivity and is defined by:

$$K_{v21} = c_{df}\pi d_n \sqrt{\frac{1}{\rho}}$$

where

c_{df} = flapper discharge coefficient, 0.85

d_n = nozzle diameter, 0.0163 inches

ρ = density of hydraulic oil, 8.14 X 10^{-5} lb-sec^2/in.^{4}

Substituting the constant values yields:

$$K_{v21} = (0.85)\,\pi\,(0.0163 \text{ in.}) \sqrt{\frac{1}{(8.14 * 10^{-5}) \dfrac{\text{lb sec}^2}{\text{in.}^4}}}$$

or in reduced form:

$$\boxed{K_{v21} = 4.82 \frac{\text{in.}^3}{\text{sec lb}^{1/2}}}$$

From the expanded baseline system block diagram in Figure 8, the term K_{v22}, a multiplying factor, is found to be:

$$K_{v22} = K_{val}\sqrt{P_s}$$

where

K_{val} = Vickers servovalve flow coefficient, 0.674

P_s = supply pressure, psi

Substituting these values back into the flapper flow equation gives:

$$Q_f = K_{v21} K_{v22} X_f$$

$$Q_f = 4.82 \frac{\text{in.}^3}{\text{sec lb}^{1/2}} (0.674) \left(\sqrt{P_s} \frac{\text{lb}^{1/2}}{\text{in.}} \right) X_f \text{ in.}$$

$$Q_f = 3.25 \sqrt{P_s} X_f \frac{\text{in.}^3}{\text{sec}}$$

VALVE SPOOL CONSTANT

The servovalve flow, Q_v, is given by the following equation:

$$Q_v = K_{v31} K_{v32} x_v$$

where

x_v = valve spool displacement, inches

The term K_{v31} is called the valve spool sensitivity and is defined by:

$$K_{v31} = c_{ds} M_o \sqrt{\frac{1}{\rho}}$$

where

c_{ds} = valve spool discharge coefficient, 0.65

M_o = metering orifice gradient, 0.9831 in.^{2}/in.

ρ = density of hydraulic oil, 8.14×10^{-5} lb-sec^2/in.^{4}

Substituting these values in the equation gives:

$$K_{v31} = (0.65) \left(0.9831 \frac{\text{in}^2}{\text{in.}} \right) \sqrt{\frac{1}{8.14 * 10^{-5} \frac{\text{lb sec}^2}{\text{in.}^4}}}$$

or in reduced form:

$$K_{v31} = 70.8 \frac{\text{in.}^3}{\text{sec lb}^{1/2}}$$

From the block diagram in Figure 8, the term K_{v32}, a multiplying factor, is given by:

$$K_{v32} = \sqrt{P_s - P_L}$$

where

P_s = supply pressure, psi

P_L = load pressure, psi

Substituting these values into the valve spool flow equation gives:

$$Q_v = K_{v31} K_{v32} X_v$$

$$Q_v = 70.8 \frac{\text{in.}^3}{\text{sec lb}^{1/2}} \sqrt{P_s - P_L} \frac{\text{lb}^{1/2}}{\text{in.}} X_v \text{ in.}$$

$$Q_v = 70.8 \sqrt{P_s - P_L}\, X_v \frac{\text{in.}^3}{\text{sec}}$$

Appendix C

HYDRAULIC ACTUATOR INTERNAL LEAKAGE FLOW COEFFICIENT CALCULATIONS

The flow rate, q, of any fluid through an orifice or nozzle can be described by the following equation:

$$Q_{le} = C_d A_{le} \sqrt{\frac{2g}{\rho}(P_s - P_L)}$$

where

Q_{le} = internal leakage flow, in.^{3}/sec

c_d = flow coefficient for square edged orifices

A_{le} = annular area, in.^{2}

ρ = density of fluid, lb/in.^{3}

P_s = supply pressure, psi

P_L = load pressure, psi

Taking the partial derivative of Q_{le} with respect to P_L gives the following equation:

$$\frac{\partial Q_{le}}{\partial P_L} = c_d A_{le} \sqrt{\frac{2g}{\rho}} \left(\frac{1}{2}\right) \frac{1}{\sqrt{P_s - P_L}}$$

Let P_s = 2,800 psi and for small perturbations let $\overline{P}_L$ = 200 psi then

$$\frac{\Delta Q_{le}}{\Delta P_L} = \frac{c_d A_{le}\sqrt{\frac{2g}{\rho}}}{2\sqrt{\frac{2800 lb}{in.^2} - P_L}}$$

If d_p = piston diameter = 1.75 in. and we assume that

Δr = piston to cylinder radial clearance = 0.002 in.

then the annular area between the piston and cylinder wall is given by:

$$A_{le} \approx \pi dp(\Delta r)$$

$$A_{le} = \pi(1.75 \text{ in.})(0.002 \text{ in.})$$

$$A_{le} = 0.011 \text{ in.}^2$$

Solving for the equivalent orifice diameter gives us d_o = 0.118 inch.

For d_o/d_p = 0.118/1.75 = 0.0676

$c_d \approx$ 0.60 (for sharp edged orifice)[1]

g = 386 in./sec^2

ρ = 0.0314 lb/in.^{3} (density of hydraulic oil)

Then the actuator internal leakage flow coefficient, K_{le}, is given by the following equation:

[1] Crane Co., Technical Paper No. 410-Flow of Fluids through Valves, Fittings, and Pipe, page A-19, Fourth edition, Chicago, Illinois, 1957.

$$K_{le} = \frac{\Delta Q_{le}}{\Delta P_L}$$

$$K_{le} = \frac{C_d A_{le} \sqrt{\dfrac{2g}{\rho}}}{2 \sqrt{(P_s - P_L)}}$$

$$K_{le} = \frac{0.60 \ (0.011 \text{ in.}^2) \sqrt{\dfrac{2 \ (386) \text{ in. } (\text{in.}^3)}{0.0314 \text{ lb sec}^2}}}{2 \sqrt{(2800 - 200) \ \dfrac{\text{lb}}{\text{in.}^2}}}$$

or

$$\boxed{K_{le} = 0.010 \ \frac{in.^3/\text{sec}}{lb/in.^2}}$$

Appendix D

CALCULATION, ω_n and ζ, FROM BASELINE STEP RESPONSES

INTRODUCTION

The damping ratio ζ can be determined from the amplitude and frequency of the peaks for the transient step response. There are two techniques for determining the damping ratio: the log decrement method and the average of peaks method. The log decrement method is based on successive amplitude ratios while the average of peaks method is based on the initial and final amplitude ratios.

The laboratory test results for the baseline system were generated by the contractor, Test Systems and Simulation, Inc. (TS&S) of Madison Heights, Michigan. The transient step responses were recorded using a Hewlett-Packard HP35665A signal analyzer and saved on a floppy disk. The values for amplitude and time were read off the plot using the Standard Data Format (SDF) viewdata software program which was provided to NCEL by TS&S.

The computer model results were generated by NCEL using the Matrix_x/System Build software programs (PC versions 7.1 and 8.0). A variable step Kutta Merson solution was performed on the baseline model using a UNISYS type 386 computer. The output data was saved to a file and imported into Lotus 123 for plotting and obtaining the amplitude and time data.

LABORATORY TEST RESULTS

The following data are from the waveform of the load position trace for a 0.5-volt square wave input and a load weight of 822 pounds. The half-amplitude of the peaks was measured from the steady state voltage which was determined to be 7.49098 volts.

Peak No. (i)	Time (sec)	Magnitude (volts)	Half-Amplitude, x (volts)
1	0.408203	7.82856	0.33758
2	0.601563	7.64669	0.15571
3	0.791016	7.54744	0.05646

Method 1: Log Decrement

The damping ratio ζ can be determined from the following equation:

$$\zeta = \frac{\delta}{2\pi}$$

where the logarithmic decrement δ is defined by the following:

$$\delta = \ln\left[\frac{x_i}{x_{i+1}}\right]$$

The experimental test data give the following equations:

$$\delta_1 = \ln\left[\frac{x_1}{x_2}\right] = \ln\left[\frac{0.33758}{0.15571}\right] = 0.7738$$

$$\delta_2 = \ln\left[\frac{x_2}{x_3}\right] = \ln\left[\frac{0.15571}{0.05646}\right] = 1.0145$$

$$\zeta_1 = \frac{\delta_1}{2\pi} = \frac{0.7738}{2\pi} = 0.1232$$

$$\zeta_2 = \frac{\delta_2}{2\pi} = \frac{1.0145}{2\pi} = 0.1615$$

The average damping ratio for the experimental data is:

$$\boxed{\zeta_{avg} = \frac{\zeta_1 + \zeta_2}{2} = \frac{0.1232 + 0.1615}{2} = 0.1423}$$

Method 2: Average of Peaks

For small values of ζ the damping ratio can also be determined using the following equations:

$$\zeta = \frac{\ln\left[\frac{x_1}{x_{1+N}}\right]}{2\pi N}$$

$$\zeta\omega_n = \frac{\ln\left[\frac{x_1}{x_{1+N}}\right]}{\tau_d}$$

where N is the number of cycles, τ_d is the time for N cycles in seconds, and ω_n is the natural frequency in rad/sec.

If N = 2 cycles and τ_d = 0.3828 seconds, then

$$\zeta = \frac{\ln\left[\frac{x_1}{x_{1+N}}\right]}{2\pi N} = \frac{\ln\left[\frac{0.33758}{0.05646}\right]}{2\pi(2)} = 0.1423$$

$$\zeta\omega_n = \frac{\ln\left[\frac{x_1}{x_{1+N}}\right]}{\tau_d} = \frac{\ln\left[\frac{0.33758}{0.05646}\right]}{0.3828\ \text{sec}} = 4.671\ \frac{\text{rad}}{\text{sec}}$$

Solving for the natural frequency, ω_n, and period, T, gives:

$$\omega_n = \frac{4.671\ \frac{\text{rad}}{\text{sec}}}{0.1423} = 32.82\ \frac{\text{rad}}{\text{sec}}$$

$$f_n = \frac{\omega_n}{2\pi} = 5.22\ \frac{\text{cycles}}{\text{sec}}$$

$$T = \frac{1}{f_n} = 0.191 \text{ sec}$$

COMPUTER MODEL RESULTS

The following results are from a Matrix_x nonlinear simulation of the baseline model using these parameters:

W_1 = load weight = 822 lb

B_2 = external load damping coefficient = 34 lb-sec/in.

P_k = forward path gain = 353.5 ma/volt

K_{le} = actuator internal leakage coefficient = 0.010 in.^{3}/sec/psi

The steady state value for the load position is 1.0000.

Peak No. (i)	Time (sec)	Magnitude (volts)	Half-Amplitude, x (volts)
1	0.12	1.3857	0.3857
2	0.33	1.0756	0.0756
3	0.53	1.0190	0.0190

Method 1: Log Decrement

The damping ratio ζ can be determined from the following equation:

$$\zeta = \frac{\delta}{2\pi}$$

where the logarithmic decrement δ is defined by the following:

$$\delta = \ln\left[\frac{x_i}{x_{i+1}}\right]$$

The computer model results gives the following equations:

$$\delta_1 = \ln\left[\frac{x_1}{x_2}\right] = \ln\left[\frac{0.3875}{0.0756}\right] = 1.6296$$

$$\delta_2 = \ln\left[\frac{x_2}{x_3}\right] = \ln\left[\frac{0.0756}{0.0190}\right] = 1.3810$$

$$\zeta_1 = \frac{\delta_1}{2\pi} = \frac{1.6296}{2\pi} = 0.2594$$

$$\zeta_2 = \frac{\delta_2}{2\pi} = \frac{1.3810}{2\pi} = 0.2198$$

The average damping ratio for the computer model data is:

$$\boxed{\zeta_{avg} = \frac{\zeta_1 + \zeta_2}{2} = \frac{0.2594 + 0.2198}{2} = 0.2396}$$

Method 2: Average of Peaks

For small values of ζ the damping ratio can also be determined using the following equations:

$$\zeta = \frac{\ln\left[\frac{x_1}{x_{1+N}}\right]}{2\pi N}$$

$$\zeta\omega_n = \frac{\ln\left[\frac{x_1}{x_{1+N}}\right]}{\tau_d}$$

where N is the number of cycles, τ_d is the time for N cycles in seconds, and ω_n is the natural frequency in rad/sec.

If N = 2 cycles and τ_d = 0.41 seconds, then:

$$\zeta = \frac{\ln\left[\dfrac{x_1}{x_{1+N}}\right]}{2\pi N} = \frac{\ln\left[\dfrac{0.3875}{0.0190}\right]}{2\pi(2)} = 0.2396$$

$$\zeta\omega_n = \frac{\ln\left[\dfrac{x_1}{x_{1+N}}\right]}{\tau_d} = \frac{\ln\left[\dfrac{0.3875}{0.0190}\right]}{0.41 \text{ sec}} = 7.3430 \ \frac{\text{rad}}{\text{sec}}$$

Solving for the natural frequency, ω_n, and period, T, gives:

$$\omega_n = \frac{7.3430 \ \dfrac{\text{rad}}{\text{sec}}}{0.2396} = 30.65 \ \frac{\text{rad}}{\text{sec}}$$

$$f_n = \frac{\omega_n}{2\pi} = 4.88 \ \frac{\text{cycles}}{\text{sec}}$$

$$T = \frac{1}{f_n} = 0.205 \text{ sec}$$

As a matter of interest the damping coefficient, b, could be determined using the following equation:

$$b = 2\zeta\omega_n m$$

where m is the mass of the load in slugs.

SUMMARY

The following table summarizes the damping ratio and frequency calculations for the baseline model for the laboratory and computer model results:

Quantity	Laboratory Model	Computer Model
Damping ratio, ζ	0.1423	0.2396
Natural frequency, ω_n, rad/sec	32.82	30.65
Natural frequency, f_n, cycles/sec	5.22	4.88
Period, T, seconds	0.191	0.205

Appendix E

DIARIES, COMPUTER SOLUTIONS, CASES 1 THROUGH 5

DIARY FILE NO. 1

ISI VARIABLE STEP KUTTA MERSON SOLUTION PERFORMED ON DEC WORKSTATION 1 MAJOR ITERATION INITIAL CONDITIONS (K_1 = 12, K_i = 1, K_2 = 0.30)

```
<>
<> cputime = clock('cpu') ; cputime = clock('cpu') ;
<>
<> load 'PDF.ASC' ;
 16 variable(s) LOADed from file: PDF.ASC
<> sim('anal/PDF') ;
Super-Block Reference Map :
  PDF
    TRANSLATIONAL MODEL
      SERVOVALVE
      FLOW CONTINUITY
      ACTUATOR LOAD
Parameters used in Super-Block : PDF
  GC       for <Gain(s)>               in  GC.1
  PK       for <Gain(s)>               in  PK.99
  K1       for <Gain(s)>               in  K1.11
  KI       for <Output Gain(s)>        in  KI.97
  OFF      for <Parameter Values>      in  OFFSET.94
  K2       for <Gain(s)>               in  K2.12
  KR       for <Gain(s)>               in  KR.93
  KFB      for <Gain(s)>               in  KFB.20
Parameters used in Super-Block : FLOW CONTINUITY
  KLE      for <Gain(s)>               in  Kle.12
Parameters used in Super-Block : ACTUATOR LOAD
  BP       for <Gain(s)>               in  Bp.22
  M1       for <Gain(s)>               in  LOAD INERTIA.1
  B1       for <Gain(s)>               in  B1.99
  W1       for <Parameter Values>      in  LOAD WEIGHT.97
  B2       for <Gain(s)>               in  B2.96

System Built with 0 error(s) and 0 warning(s).
Use  SIM('IALG') to set the integration algorithm
<> sim('noclock,nomessage') ;
<> sim('ialg=5') ;
<>
<> PAR_HIS = [K1;KI;K2] ;
<> t = [0:1e-2:0.5]' ;
<> y = sim(t,ones(t)); plot(t,y(:,1)) ;
<> COST_HIS = max(abs(y(100*0.15+1:51,1)-1)) ;
<> Y_HIS = y(:,1) ;
<> save 'history.dat' PAR_HIS COST_HIS Y_HIS ;
<>
<> define 'cost_opt.udf'

 OUT =COST(VP,ITER)
```

< > vP = [k1;ki;k2] ;
< > [vp vcost] = optimize(vP,[-200 200],[0 1 50 0 0]) ;
J = **This is the cost function which is the amount of overshoot.**
0.1200

CONSTRAINT = **The constraint is the actuator current which is limited to 200 ma.**
35.4463
Major Iteration 1
Minor iteration 1
3 variable(s) LOADed from file: history.dat
Minor iteration 2
3 variable(s) LOADed from file: history.dat
Minor iteration 3
3 variable(s) LOADed from file: history.dat
Minor iteration 4
3 variable(s) LOADed from file: history.dat
Minor iteration 5
3 variable(s) LOADed from file: history.dat
Minor iteration 6
3 variable(s) LOADed from file: history.dat
Minor iteration 7
3 variable(s) LOADed from file: history.dat
Minor iteration 8
3 variable(s) LOADed from file: history.dat
Minor iteration 9
3 variable(s) LOADed from file: history.dat
Minor iteration 10
3 variable(s) LOADed from file: history.dat
Minor iteration 11
3 variable(s) LOADed from file: history.dat
Minor iteration 12
3 variable(s) LOADed from file: history.dat
Minor iteration 13
3 variable(s) LOADed from file: history.dat
Minor iteration 14
3 variable(s) LOADed from file: history.dat

ANS = **This is an optimal set of the parameters K_1, K_i, and K_2 which produces an overshoot**
12.5345 **of 0.0099 or 0.99%. The actuator current of 42.0003 ma is well below the saturation limit of 200 ma.**
0.8681
-0.4483
3 variable(s) LOADed from file: history.dat
J =

0.0099
CONSTRAINT =

42.0003
OPTIMIZE--> Exiting after maximum number of iterations
Tolerance not achieved
< >
< > cputime = clock('cpu') - cputime ,

```
CPUTIME   =    The CPUTIME shown here is about 3.5 minutes and is how long it took the ISI DEC
               WORKSTATION to solve the optimization problem.
208.8300
  no flops
<>
<> load 'history.dat'
 3 variable(s) LOADed from file: history.dat
<>
<> COST_HIS ,     The COST_HIS variable stores the amount of
               overshoot for each major and minor iteration.
 COST_HIS  =

   0.1200
   0.0989
   0.0760
   0.0606
   0.0604
   0.0578
   0.0477
   0.0374
   0.0270
   0.0165
   0.0130
   0.0101
   0.0099
   0.0099
   0.0099
  no flops
<> PAR_HIS,     This is a list of the parameter updates after each
             major and minor iteration.
 PAR_HIS  =

  Columns    1 thru    5
 12.0000   12.0219   12.0432   12.0630   12.0589
  1.0000    0.9511    0.8992    0.8462    0.8602
  0.3000    0.1671    0.0264   -0.1171   -0.0800

  Columns    6 thru   10
 12.0827   12.1797   12.2771   12.3744   12.4708
  0.8630    0.8635    0.8640    0.8643    0.8646
 -0.0921   -0.1683   -0.2456   -0.3237   -0.4019

  Columns   11 thru   15
 12.5017   12.5323   12.5375   12.5345   12.5345
  0.8660    0.8663    0.8698    0.8681    0.8681
 -0.4244   -0.4493   -0.4468   -0.4483   -0.4483
  no flops
<> diary(0)
```

DATE OF RUN: 4-2-92

DIARY FILE NO. 2

ISI FIXED STEP KUTTA MERSON SOLUTION PERFORMED ON DEC WORKSTATION 1 MAJOR ITERATION INITIAL CONDITIONS ($K_1 = 12$, $K_i = 1$, $K_2 = 0.30$)

```
< >
< > cputime = clock('cpu') ,

 CPUTIME   =    This is the initial clock setting before the simulation is performed.
 650.1100
   no flops
< >
< > load 'PDF.ASC' ;
ACTUATOR LOAD replaced.
FLOW CONTINUITY replaced.
PDFOL replaced.
SERVOVALVE replaced.
SVALVE replaced.
TRANSLATIONAL MODEL replaced.
PDFCL replaced.
PDF replaced.
 16 variable(s) LOADed from file: PDF.ASC
< > sim('anal/PDF') ; sim('noclock,nomessage') ;
Super-Block Reference Map :
  PDF
    TRANSLATIONAL MODEL
      SERVOVALVE
      FLOW CONTINUITY
      ACTUATOR LOAD
Parameters used in Super-Block : PDF
  GC      for <Gain(s)>             in  GC.1
  PK      for <Gain(s)>             in  PK.99
  K1      for <Gain(s)>             in  K1.11
  KI      for <Output Gain(s)>      in  KI.97
  OFF     for <Parameter Values>    in  OFFSET.94
  K2      for <Gain(s)>             in  K2.12
  KR      for <Gain(s)>             in  KR.93
  KFB     for <Gain(s)>             in  KFB.20
Parameters used in Super-Block : FLOW CONTINUITY
  KLE     for <Gain(s)>             in  Kle.12
Parameters used in Super-Block : ACTUATOR LOAD
  BP      for <Gain(s)>             in  Bp.22
  M1      for <Gain(s)>             in  LOAD INERTIA.1
  B1      for <Gain(s)>             in  B1.99
  W1      for <Parameter Values>    in  LOAD WEIGHT.97
  B2      for <Gain(s)>             in  B2.96

System Built with 0 error(s) and 0 warning(s).
Use  SIM('IALG') to set the integration algorithm
< > sim('ialg=4,noclock,nomessage,hold') ;
< > t = [0:1e-3:0.5]' ;
```

```
<> y = sim(t,ones(t)); plot(t,y(:,1)) ;
<> define 'fixed.udf'
 OUT =COST(VP,ITER)
<> vP = [k1;ki;k2] ;
<> [vp vcost] = optimize(vP,0,[0 1 10 0 0]) ;

 J         =

   0.1202                          These are the parameter updates
 Major Iteration 1                  after each minor iteration.
 Minor iteration 1
 +0.9351259588195247,-1                Amount of overshoot
 +0.3202720312580238,+2                 Actuator currrent, ma
 +0.1202535944990714,+2                 Update for K1
 +0.8753039824460357,+0                 Update for Ki
 +0.1961277583028945,+0                 Update for K2
 Minor iteration 2
 +0.8216736710740191,-1
 +0.3666203549519677,+2
 +0.1204132344734976,+2
 +0.9345084942951297,+0
 +0.4393893091109394,-1
 Minor iteration 3
 +0.7574368399984909,-1
 +0.5227928727972588,+2
 +0.1213183707063297,+2
 +0.9487764176262366,+0
 -0.6525978426125127,+0
 Minor iteration 4
 +0.3157173673958579,-1
 +0.4137971380806804,+2
 +0.1209134636765469,+2
 +0.9061199371898338,+0
 -0.2870566154822436,+0
 Minor iteration 5
 +0.7651953899197305,-1
 +0.3814949807614759,+2
 +0.1217869378076218,+2
 +0.6794175555724746,+0
 -0.6342498204199315,+0
 Minor iteration 6
 +0.1658920740300962,-1
 +0.4216733452358816,+2
 +0.1211599585006388,+2
 +0.8692940130086562,+0
 -0.4042179266618151,+0
 Minor iteration 7
 +0.1968479500867381,-1
 +0.4388544691370437,+2
 +0.1213167484122660,+2
 +0.8571836287655413,+0
 -0.5044237899612196,+C

 Minor iteration 8
```

+0.1062163560187978,-1	**Amount of overshoot**
+0.3202720312580238,+2	**Actuator currrent, ma**
+0.1202535944990714,+2	**Update for K_1**
+0.8753039824460357,+0	**Update for K_i**
+0.1961277583028945,+0	**Update for K_2**

Minor iteration 2
+0.4206569680628394,+2
+0.1212641636154921,+2
+0.8476913442391802,+0
-0.4452853787275462,+0
Minor iteration 9
+0.1467887661753586,-1
+0.4241778642333120,+2
+0.1213759778102861,+2
+0.8288985511921152,+0
-0.4998455639208904,+0
Minor iteration 10
+0.1869882994169758,-1
+0.4254332151489956,+2
+0.1214030051405884,+2
+0.8226133258359508,+0
-0.5179547643285506,+0
Updated parameters

ANS =

12.1325
0.8373
-0.4754
+0.9448780832438162,-2
+0.4225619602849151,+2
+0.1213252541146203,+2
+0.8373148463481667,+0
-0.4754023021369891,+0

J =

0.0094
OPTIMIZE--> Exiting after maximum number of iterations
Tolerance not achieved
< > cputime = clock('cpu') ,

CPUTIME = **This time must be subtracted from the initial clock setting to get the actual computational**
972.8400 **time for this solution.**
no flops

ACTUAL CPUTIME = CPUTIME - INITIAL TIME
= 972.8400 - 650.1100
= 322.73 seconds
= 5 minutes and 22 seconds

< > diary(0) **NOTE: The type 486 PC solution for the identical optimization problem was approximately 47 minutes.**

DATE OF RUN: 4-7-92

DIARY FILE NO. 3

ISI FIXED STEP KUTTA MERSON SOLUTION PERFORMED ON DEC WORKSTATION 4 MAJOR ITERATIONS INITIAL CONDITIONS (K_1 = 12, K_i = 1.0, K2= 0.30)

```
< >
< > cputime = clock('cpu') ; cputime = clock('cpu') ;
< >
< > exec('opt_old2.exe',1) ;
< > // mws21x DECstation 5000/120 : [0 25 50 0 0] exec in 590s
< >
< > load 'PDF.ASC' ;
 16 variable(s) LOADed from file: PDF.ASC
< > sim('anal/PDF') ;
Super-Block Reference Map :
  PDF
    TRANSLATIONAL MODEL
      SERVOVALVE
      FLOW CONTINUITY
      ACTUATOR LOAD
Parameters used in Super-Block : PDF
  GC      for <Gain(s)>             in  GC.1
  PK      for <Gain(s)>             in  PK.99
  K1      for <Gain(s)>             in  K1.11
  KI      for <Output Gain(s)>      in  KI.97
  OFF     for <Parameter Values>    in  OFFSET.94
  K2      for <Gain(s)>             in  K2.12
  KR      for <Gain(s)>             in  KR.93
  KFB     for <Gain(s)>             in  KFB.20
Parameters used in Super-Block : FLOW CONTINUITY
  KLE     for <Gain(s)>             in  Kle.12
Parameters used in Super-Block : ACTUATOR LOAD
  BP      for <Gain(s)>             in  Bp.22
  M1      for <Gain(s)>             in  LOAD INERTIA.1
  B1      for <Gain(s)>             in  B1.99
  W1      for <Parameter Values>    in  LOAD WEIGHT.97
  B2      for <Gain(s)>             in  B2.96

System Built with 0 error(s) and 0 warning(s).
Use  SIM('IALG') to set the integration algorithm
< >
< > sim('noclock,nomessage') ;
< >
< > define 'cost_old.udf'

 OUT =COST(VP,ITER)
< > t = [0:1e-3:0.5]' ; // 500ms window
< > //K1 = 7.6 ; KI = 0.47 ; K2 = -0.75 ;
< > sim('ialg=4') ; // Fixed KM
< > y = sim(t,ones(t)) ; plot(t,y(:,1)) ;
< > vP = [k1;ki;k2] ;
```

```
< > run = "[vp vcost] = optimize(vP,[-200 200],[0 25 50 0 0]) ;" ;
< > PAR = vP ;
< >
< > ]run[ ;

J          =

  0.1202

CONSTRAINT =

 35.4452
Major Iteration 1
Minor iteration 1
Minor iteration 2
Minor iteration 3
Minor iteration 4
Minor iteration 5
Minor iteration 6
Minor iteration 7
Minor iteration 8
Minor iteration 9
Minor iteration 10
Minor iteration 11
Updated parameters

ANS        =

 12.5641
  0.8699
 -0.4456

J          =

  0.0099

CONSTRAINT =

 42.1336
Major Iteration 2
Minor iteration 1
Updated parameters

ANS        =
 12.5646
  0.8699
 -0.4460
J          =

  0.0099

CONSTRAINT =

 42.1383
```

```
Major Iteration 3
Minor iteration 1
Updated parameters

 ANS        =

  12.5649
   0.8698
  -0.4462

 J          =

   0.0099

 CONSTRAINT =

  42.1407
Major Iteration 4
Minor iteration 1
Updated parameters

 ANS=          NOTE: The parameter set did not change very much during the additional major iterations.

  12.5649
   0.8698
  -0.4462

 J          =

   0.0099

 CONSTRAINT =

  42.1409
OPTIMIZE--> Completed in 4  iterations
< >
< > PAR = [ PAR , vP ] ;
< >
< > save 'opt_old2.dat' PAR vcost
< >
< >
< > RETURN
< >
< > cputime = clock('cpu') - cputime ,


 CPUTIME    =           The CPUTIME for this solution was about 10 minutes.
 589.8800
   no flops
< > diary(0)
```

DATE OF RUN: 3-30-92

DIARY FILE NO. 4

NCEL VARIABLE STEP KUTTA MERSON SOLUTION PERFORMED ON UNISYS 386/25 MHZ COMPUTER INITIAL CONDITIONS (K_1 = 8.0, K_i = 0.75, K_2 = 0.20)

```
16 variable(s) LOADed from file: PDF1
SuperBlock Reference Map :
  PDF
    TRANSLATIONAL MODEL
      SERVOVALVE
      FLOW CONTINUITY
      ACTUATOR LOAD
Parameters used in SuperBlock : PDF
  GC     for <Gain(s)>              in GC.1
  PK     for <Gain(s)>              in PK.99
  K1     for <Gain(s)>              in K1.11
  KI     for <Output Gain(s)>       in KI.97
  OFF    for <Parameter Values>     in OFFSET.94
  K2     for <Gain(s)>              in K2.12
  KR     for <Gain(s)>              in KR.93
  KFB    for <Gain(s)>              in KFB.20
Parameters used in SuperBlock : FLOW CONTINUITY
  KLE    for <Gain(s)>              in Kle.12
Parameters used in SuperBlock : ACTUATOR LOAD
  BP     for <Gain(s)>              in Bp.22
  M1     for <Gain(s)>              in LOAD INERTIA.1
  B1     for <Gain(s)>              in B1.99
  W1     for <Parameter Values>     in LOAD WEIGHT.97
  B2     for <Gain(s)>              in B2.96

System Built with 0 error(s) and 0 warning(s).
Use  SIM('IALG') to set the integration algorithm

 OUT =COST(VP,ITER)

 J        =

   .1977

 CONSTRAINT =

  34.3488
Major Iteration 1
Minor iteration 1
  3 variable(s) LOADed from file: HISTORY.DAT
Minor iteration 2
  3 variable(s) LOADed from file: HISTORY.DAT
Minor iteration 3
  3 variable(s) LOADed from file: HISTORY.DAT
Minor iteration 4
  3 variable(s) LOADed from file: HISTORY.DAT
Minor iteration 5
  3 variable(s) LOADed from file: HISTORY.DAT
```

Updated parameters

ANS =
8.1321
.5033
-.5256
3 variable(s) LOADed from file: HISTORY.DAT

J =

.0464

CONSTRAINT =

34.9403
Major Iteration 2
Minor iteration 1
3 variable(s) LOADed from file: HISTORY.DAT
Updated parameters

ANS =

8.1311
.5125
-.5043
3 variable(s) LOADed from file: HISTORY.DAT

J =

.0456

CONSTRAINT =

34.9502
Major Iteration 3
Minor iteration 1
3 variable(s) LOADed from file: HISTORY.DAT
Updated parameters

ANS =

8.1340
.5131
-.5035
3 variable(s) LOADed from file: HISTORY.DAT

J =

.0443
CONSTRAINT =

34.9599
OPTIMIZE--> Completed in 3 iterations

CPUTIME = **The CPUTIME is about 1 hour and 50 minutes.**

6.6487D+03
3 variable(s) LOADed from file: HISTORY.DAT

ANS =

.1977
.1699
.1359
.0935
.0471
.0464
.0464
.0456
.0456
.0443
.0443
no flops

PAR_HIS =

Columns 1 thru 6

8.0000	8.0331	8.0660	8.0984	8.1291	8.1321
0.7500	.6989	.6422	.5786	.5064	.5033
.2000	.0491	-.1178	-.3044	-.5155	-.5256

Columns 7 thru 11

8.1321	8.1311	8.1311	8.1340	8.1340
0.5033	0.5125	.5125	.5131	.5131
-.5256	-.5043	-.5043	-.5035	-.5035

no flops

DATE OF RUN: 4-7-92

DIARY FILE NO. 5

NCEL FIXED STEP KUTTA MERSON SOLUTION PERFORMED ON UNISYS 386/25 MHZ COMPUTER INITIAL CONDITIONS (K_1 = 8.0, K_i = 0.75, K_2 = 0.20)

```
 16 variable(s) LOADed from file: PDF1
SuperBlock Reference Map :
  PDF
    TRANSLATIONAL MODEL
      SERVOVALVE
      FLOW CONTINUITY
      ACTUATOR LOAD
Parameters used in SuperBlock : PDF
  GC      for <Gain(s)>              in  GC.1
  PK      for <Gain(s)>              in  PK.99
  K1      for <Gain(s)>              in  K1.11
  KI      for <Output Gain(s)>       in  KI.97
  OFF     for <Parameter Values>     in  OFFSET.94
  K2      for <Gain(s)>              in  K2.12
  KR      for <Gain(s)>              in  KR.93
  KFB     for <Gain(s)>              in  KFB.20
Parameters used in SuperBlock : FLOW CONTINUITY
  KLE     for <Gain(s)>              in  Kle.12
Parameters used in SuperBlock : ACTUATOR LOAD
  BP      for <Gain(s)>              in  Bp.22
  M1      for <Gain(s)>              in  LOAD INERTIA.1
  B1      for <Gain(s)>              in  B1.99
  W1      for <Parameter Values>     in  LOAD WEIGHT.97
  B2      for <Gain(s)>              in  B2.96

System Built with 0 error(s) and 0 warning(s).
Use  SIM('IALG') to set the integration algorithm

 OUT =COST(VP,ITER)

 J        =

   .1977
Major Iteration 1
Minor iteration 1
  3 variable(s) LOADed from file: HISTORY.DAT
Minor iteration 2
  3 variable(s) LOADed from file: HISTORY.DAT
Updated parameters

 ANS      =

  8.0306
   .5886
   .1293
  3 variable(s) LOADed from file: HISTORY.DAT
```

```
J        =

  .0227
Major Iteration 2
Minor iteration 1
 3 variable(s) LOADed from file: HISTORY.DAT
Updated parameters

 ANS     =

  8.0306
   .5886
   .1293
 3 variable(s) LOADed from file: HISTORY.DAT

J        =

  .0227
OPTIMIZE--> Completed in 2  iterations

 CPUTIME   =
  8.2538D+03
 3 variable(s) LOADed from file: HISTORY.DAT

 ANS     =

   .1977
   .1483
   .1280
   .0227
   .1280
   .0227
  no flops

 PAR_HIS   =

  8.0000   8.0408   8.0704   8.0306   8.0704   8.0306
   .7500    .5350    .2765    .5886    .2761    .5886
   .2000    .1058    .0485    .1293    .0485    .1293
  no flops
```

The CPUTIME for this solution is about 2 hr and 17 minutes.

DATE OF RUN: 4-8-92

Appendix F

PRODUCTIVITY ANALYSIS, OPTIMAL TUNING

INTRODUCTION

The main purpose of optimal tuning heavy equipment motion controllers in this study is to improve productivity. Reducing the overshoot and settling time are some of the ways productivity can be increased. The productivity can be expressed in terms of a frequency of operation. If the number of operational cycles per second can be increased, then the amount of work done will also increase for the same time period. Another way of looking at this is to reduce the cycle time for a particular operation since frequency is the reciprocal of the period.

An analysis of the optimally tuned system's productivity compared to the baseline and nonoptimal PDF models was performed. A summary of these results along with other system performance data is shown in Tables 8 and 9 in the main text of this report. The analysis which follows shows the development of the equations used.

BACKGROUND

The total cycle time can be expressed by the following equation:

$$T_t = T_y + T_b$$

where

T_t = total cycle time, sec/cycle
T_y = yaw cycle time, sec/cycle
T_b = balance of cycle time, sec/cycle

There are three principal modes of motion: yaw, roll, and pitch. The yaw cycle time is proportional to the settling time and T_b is the time needed for the pitch, roll, and any other modes of motion. The control system studied in this report was dynamically analogous to a backhoe boom operating in the yaw mode. The other directions of motion, namely the pitch and roll, were not examined. The productivity of a system can be expressed as the reciprocal of the total cycle time or:

$$P = \frac{1}{T_t}$$

where

P = the system productivity, cycles/sec
T_t = the total cycle time, sec/cycle

Therefore in order to improve productivity, the yaw cycle time needs to be reduced. The optimization of this control system involved integrating the pseudo-derivative feedback or PDF controller algorithm into a baseline system, which contained only simple proportional feedback, to minimize the overshoot and settling time. The PDF algorithm consisted of three adjustable parameters, K_1, K_i, and K_2, which represented proportional, integral, and derivative gain terms. A nonoptimal set of PDF parameters was shown to reduce the overshoot and settling time compared to a baseline system (see Figure 29 in the main text). The optimization of the PDF parameter sets using computer techniques provided further improvement in system performance. The graphs in Figures 35 through 41 and the data in Tables 3 through 8 indicate the significant improvements available from optimal tuning.

DERIVATION OF PRODUCTIVITY EQUATIONS

The productivity associated with an original or nonoptimal set of parameters can be expressed as:

$$P_o = \frac{1}{T_{to}} = \frac{1}{T_{yo} + T_{bo}}$$

where

P_o = original productivity, cycles/sec
T_{to} = original cycle time, sec/cycle
T_{yo} = original yaw cycle time, sec/cycle
T_{bo} = balance of cycle time, sec/cycle

The yaw cycle time can also be expressed by the following:

$$T_{yo} = aT_{to}$$

where a is the original yaw duty cycle in percent. Solving for T_{to} and substituting into the time balance equation gives:

$$\frac{T_{yc}}{a} = T_{yo} + T_{bo}$$

Rearranging terms gives us:

$$T_{bo} = \left[\frac{1}{a} - 1\right] T_{yo}$$

Substituting this expression into the productivity equation gives:

$$P_o = \frac{1}{T_{yo} + \left[\frac{1}{a} - 1\right] T_{yo}}$$

or in reduced form:

$$\boxed{P_o = \frac{a}{T_{yo}}}$$

Similarly if we apply the same logic to the productivity associated with the final parameter set we get:

$$\boxed{P_f = \frac{b}{T_{yf}}}$$

where b is the final yaw duty cycle in percent and T_{yf} is the final yaw cycle time in seconds/cycle. Since b is the final yaw duty cycle which is normally an unknown quantity, it is desirable to solve for b in terms of a:

$$a = \frac{T_{yo}}{T_{to}} = \frac{T_{yo}}{T_{yo} + T_{bo}}$$

$$b = \frac{T_{yf}}{T_{tf}} = \frac{T_{yf}}{T_{yf} + T_{bf}}$$

It is reasonable to assume that T_{bo} and T_{bf} are equal hence,

$$b = \frac{T_{yf}}{T_{yf} + T_{bo}} = \frac{T_{yf}}{T_{yf} + \left[\frac{1}{a} - 1\right] T_{yo}}$$

Dividing through by T_{yf} and letting $\alpha = T_{yo}/T_{yf}$ gives:

$$b = \frac{1}{1 + \left[\frac{1 - a}{a}\right]\alpha}$$

or in reduced form:

$$\boxed{b = \frac{a}{a\,[1 - \alpha] + \alpha}}$$

YAW DUTY CYCLE IMPROVEMENT

The improvement of the yaw duty cycle time can be expressed as:

$$\Delta T_{dc} = \frac{b - a}{a} = \frac{\left[\frac{a}{a\,[1 - \alpha] + \alpha}\right] - a}{a}$$

or in reduced form:

$$\boxed{\Delta T_{dc} = \frac{1}{a\,[1 - \alpha] + \alpha} - 1}$$

If we let T_{yf} = optimal PDF yaw cycle time, then the improvement in the final yaw duty cycle time from the baseline or the nonoptimal PDF models can be determined by using the following values for T_{yo}:

Improvement from baseline, let T_{yo} = baseline yaw cycle time

Improvement from nonoptimal PDF, let T_{yo} = nonoptimal PDF yaw cycle time

The percentage improvements in the cycle time shown in Table 8 were determined by multiplying the values obtained for ΔT_{dc} by 100, and were based on an assumed initial yaw duty cycle of 40 percent.

OVERALL PRODUCTIVITY GAIN

The overall productivity gain can be expressed by the following equation:

$$\Delta P_g = \frac{P_f - P_o}{P_o} = \frac{\frac{b}{T_{yf}} - \frac{a}{T_{yo}}}{\frac{a}{T_{yo}}}$$

Substituting for b gives:

$$\Delta P_g = \frac{\frac{\frac{a}{a\,[1 - \alpha] + \alpha}}{T_{yf}} - \frac{a}{T_{yo}}}{\frac{a}{T_{yo}}}$$

and since

$$\alpha = \frac{T_{yo}}{T_{yf}}$$

then through algebraic simplification, the reduced form of the equation becomes:

$$\boxed{\Delta P_g = \left[\frac{\alpha}{a\,(1 - \alpha) + \alpha}\right] - 1}$$

If we let T_{yf} = optimal PDF yaw cycle time then the overall improvement in productivity from the baseline or the nonoptimal PDF models can be determined by using the following values for T_{yo}:

Improvement from baseline, let T_{yo} = baseline yaw cycle time

Improvement from nonoptimal PDF, let T_{yo} = nonoptimal PDF yaw cycle time

The percentage improvements in the productivity shown in Table 8 were determined by multiplying the values obtained for ΔP_g by 100, and were based on an assumed value initial yaw duty cycle of 40 percent.

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